Structural analysis and condition monitoring of grinding mills: A case study

Licentiate thesis

by

Filip Berglund

Division of Operation and Maintenance Engineering
Luleå University of Technology
Preface

The research work presented in this thesis has been carried out at the Division of Operation and Maintenance Engineering at Luleå University of Technology over the time span between 2010 and 2012.

First I would like to thank my supervisor Professor, Uday Kumar, for having full faith in me during my research. I would also like to thank Dr Aditya Parida, who was my supervisor during the first year. Furthermore, I would like to acknowledge with thanks the financial support from VINNOVA, LKAB and Ringhals. Thanks are also extended to Dr Tao Xin for reading and giving comments on this thesis. Finally, I would like to thank Professor Jan Lundberg and Dr Johan Tillberg for their support during this work.

To my family.

Filip Berglund, 19/9/2012, Luleå
Abstract

Grinding mills are large rotating cylindrical steel vessels used to grind ore and minerals into finer particles. The mills are important parts of the mineral enrichment process and the grinding is the last step of the comminution process, where the particle size is reduced by a combination of abrasion and impact.

The rotation of the mill under loaded conditions can result in fatigue cracks. Fatigue cracks and associated failures have been identified as a major problem in mineral processing plants. The cracks lead to unpredicted and unplanned production stoppages for inspections and for repair and replacement of the cracked mill parts. This leads to increasing costs due to production loss, additional man-hours and spare parts.

The purpose of the research presented in this licentiate thesis was to calculate the structural strains, stresses, displacements, etc. in grinding mills in operation, to prevent overloading, to calculate crack propagation speeds and critical crack lengths, and to develop new improved mills that would withstand the current loading. This research has also aimed to propose, develop and test methods for the detection and monitoring of fatigue cracks in mills during operation, in order to facilitate optimal maintenance decision-making based on current crack sizes.

The performed research is a case study of the secondary pebble mills of LKAB, a mining company in northern Sweden. The mills are situated inside dressing plants KA1 and KA2 in Kiruna.

To achieve the goals, a number of crack detection and monitoring methods were investigated and evaluated as to their ability to find and monitor fatigue cracks on the running mills. Measurements with wireless strain measurement equipment, infrared thermography and crack propagation sensors were performed on the mills in operation.

A finite element model of a mill was developed to calculate the strains and stresses in the mill at any position in the mill and for any loading condition. A variety of spatial discretizations, boundary conditions, material properties and loading alternatives were considered to simulate the behaviour of the real mill in the best possible way. To calculate the loading on the mills in operation, a mathematical model and computer software were developed to calculate the charge configuration, as well as the loading and the magnitude and distribution of the forces acting on the mill in operation. Using the finite element model and the computer software, the global displacement field of the entire mill structure was calculated using quasi-static loading for different inputs of the charge and process parameters.

To verify the finite element results, the measured strain ranges for one complete rotation of the mill were compared with the corresponding calculated ones. The numerical results were also verified with logged process data, such as bearing reaction forces. One conclusion, based on the comparisons, is that the developed finite element model and the developed software tools can be considered useful for engineering applications.
The developed software tools, together with the finite element model, make it possible to calculate the global displacement field of the entire mill structure for any situation. This is achieved by inputting the desired process data and charge parameters into the software, calculating the loads and force distributions, exporting them to the finite element model, and running the simulation. From the global displacement field, strains, stresses, reaction forces, displacements, etc. can be calculated with standard routines for any position in the mill.

The performed research work gives a deeper understanding of the field of structural analysis and load calculation of grinding mills in operation. The complexity of modelling the behaviour of mills in operation is high. Consequently, it is difficult to obtain accurate estimations of crack propagation speeds and critical crack sizes based on the calculated stresses.

It has been found that strain measurements, with strain gauges attached to the mill mandrel, can be used to detect and monitor larger circumferential cracks near the flanges in the mill in operation, since the measured strain ranges increase with the crack size.

It has further been found that infrared thermography can be used as a method to indicate cracks without stopping the mill, as the increased thermal gradient around the cracks can be detected by a special type of thermal instrument.

Crack propagation sensors have proven to be ideal for high-precision online monitoring of the crack propagation of smaller cracks at the corners of the manholes in the mill.

Finally, it has been found that strain measurement is a useful method not only to verify finite element results and to detect and monitor cracks, but also to prevent overloading of the mill and to estimate charge features such as the filling level, the charge shape and the position of the charge circumferentially inside the mill during operation.

**Keywords:** Condition monitoring, Finite element analysis, Loading analysis, Wireless strain measurements, Programming, Computer software, Charge analysis, Structural analysis, Grinding mill, Fatigue crack, Infrared thermography
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Paper B

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**Introduction**

Grinding mills are large rotating cylindrical steel vessels used to grind particles into finer pieces. The mills are important parts of the mineral enrichment process and the performed grinding is the last step of the comminution process, where the particle size is reduced by a combination of impact and abrasion. Grinding mills exist in a large variety of sizes and models for different types of grinding.

The mills investigated in the research work presented in this thesis are autogenous pebble mills of the discharging type which utilize the ore itself as grinding media. The investigated mills are situated at room temperature inside dressing plants KA1 and KA2 at the Swedish mining company LKAB (Loussavaara-Kirunavaara Aktiebolag) in Kiruna.

Two different types of pebble mills have been studied, each of which has a different length, diameter, installed power and bearing support. One is supported with hydrostatic pressure bearings and the other with roller bearings.

The investigated mills are located as secondary mills (after the primary mills) in a production line of two mills. The mills typically consist of 6-7 cylindrical parts which are assembled together with bolts. The typical construction of the mills can be divided into three main parts: the mandrel, the inlet head and the outlet head. The heads are the circular plates attached to each end of the mandrel, see Figure 6.

The iron ore material, together with a large amount of water, enters the mill at the inlet end. The ore is crushed and ground to a fine powder inside the mill, in a process where larger ore lumps are grinding smaller ones. At the outlet end, ore with a small enough lump size passes through a grate and is lifted out of the mill via a discharge wheel. The ore is then poured into a trommel sieve, beyond which the finest powder grades are further enriched and processed. The inside surface of the mill is covered with magnetic linings, mainly for protection of the mill surface.

The grinding charge inside the mills consists of a mix of larger and finer iron ore pieces and a large amount of water. As the mill rotates, the charge moves inside the mill and extends pressure and friction on the mill. Due to constant changes in the process and charge parameters, e.g. the engine power, filling level, charge mass, ore density, the amount of water in the charge, etc., the charge and the loading on the mill vary over time during production.

The mills are in service all around the clock 365 days a year, except for one or two maintenance stoppages for service and lining replacement, which usually last around one week each. The expected lifetime for certain mills is about 50 years.
Statement of the problem

As the mills rotate under loaded conditions, the material undergoes fatigue and each rotation of the mill is equivalent to one fatigue cycle. At a typical rotational speed of 12.75 RPM the mills undergo about 6.5 million fatigue cycles a year, which means that the mills are subjected to high cycle fatigue.

Fatigue cracks have recently become a frequent problem in the mills at LKAB. The cracks put the mills out of operation a long time before the end of their expected lifetime. Propagating cracks of up to 2 m are not unusual in mills which are only about 10-20 years of age.

Cracks often grow circumferentially near flanges at the inlet or outlet end of the mill and in connection with fillet or butt welds, see Figure 1. The directions of the cracks are often maintained during propagation, but several smaller cracks propagate in a spreading pattern at each crack end, which makes it difficult to measure and predict the true extension of the crack.

Other common areas for fatigue cracks are the corners of the manholes. The manholes are squared holes in the mandrel which have raided corners, are covered with plates, and are used for inspections inside the mill, see Figure 2. Cracks at these locations have usually only one crack tip and propagate in a very straight and predictable manner.

Upcoming cracks propagate at an increasing rate with time and reach at a certain point a critical size leading to failure and breakdown of the mill. The crack propagation speed depends on the material properties of the mill and the stress intensity range at the crack tips. The stress intensity range is a function of the crack length itself, the geometry at the crack location, the nominal stress range at the position of the crack, and whether or not the crack is located in a heat-affected zone with weld residual stresses.
In general, the cracks initiate from high stress concentrations near welded joints in the mill. Welds are sensitive to fatigue due to defects and residual stresses in the heat-affected zone (Eriksson, 2009). Welds usually contain notches, voids and defects which are typical initiating points for cracks. Additional factors adding to the risk of crack initiation are the corrosive and wet environment inside the mill and the large size of the machine. The large machine size increases the risk for defects in the material and increases the relative sharpness of notches.

Today cracks are discovered during manual inspections during maintenance stoppages. Cracks are often found at a stage when a certain crack size has been reached. Maintenance actions are carried out whereby the crack is photographed, measured and inspected by ultrasonic equipment. If the crack is considered to be small enough, the mill is started again immediately, otherwise the mill is stopped until repair or replacement of the broken part has taken place.

The cracks are often repaired quickly after being found through welding, an operation which usually takes around one week and includes risks. The welding can lead to thermal deformations of the mill, with the geometry and roundness of the mill changing and puts the mill out of operation. The welding can also introduce flaws and stress raisers which make the situation worse.

Circumferential cracks near flanges are often repaired with triangular plates which are welded over the crack in order to close the crack, see Figure 3. The repair work through welding is usually successful and reduces the propagation speed of the crack. This solution is, however, only temporary and to solve the problem permanently the cracked part must be replaced.
The replacement of a cracked mill part requires that a new spare part should be in stock. The replacement of cracked parts can take any length of time from one week to one month, depending on the mill type and crack position. Figure 4 shows the replacement of a part of the outlet head on a mill supported by hydrostatic bearings. The spare parts are large, often with a diameter of 6.22 - 6.98 m and a length of 2.74 - 3.1 m and with masses of 20-50 tons. The delivery time for a new part is about one year from the ordering date.

Unpredicted cracks lead to additional costs due to repair work, production stoppages, spare parts, replacement, etc. and are a major concern for the mining company.

**Purpose and objectives**

Unpredicted mill breakdowns lead to production stoppages and high costs. To be able to plan repair work and replacement in an adequate and optimal way, it is necessary to know how long the cracked mill can be safely operated before failure. This can help the management to decide whether a crack should be repaired immediately, with production loss as a result, or during the next scheduled maintenance stoppage. The breakdown of a machine is fatal and must be avoided.
at all costs, and therefore maintenance decisions are usually conservative and high accuracy and reliability are important in predictions of the remaining useful life of cracked mills.

High reliability of predictions of the crack growth speed and the time to failure leads to better maintenance decisions and saved costs due to more optimal spare part handling, more machine time and reduced repair work. For example, if the time to failure is predicted to be two years from the date of crack detection, no immediate repair work at all needs to be undertaken and normal production can be maintained without interruptions; the cracked part can then be replaced in a planned manner during a regular maintenance stoppage two years later. Without information on the time to failure, the crack needs to be repaired or the part replaced as soon as possible after detection due to the uncertainty of the failure time and the need to be on the safe side. This leads to a 1-2 week production loss due to replacement and repair work, extra man-hour costs and inadequate use of spare parts.

Proper estimations of the remaining useful life are necessary to plan repair work and replacements in an optimal and cost-saving manner. However, to solve problems related to cracks permanently, one of the following three actions, a combination of them, or all of them need to be undertaken:

1. Improvement of future mill designs with the aim of lowering the stresses in mills in operation to make them more resistant to fatigue and avoid cracks.
2. Reduction of the loading on the mills, which means a lowered charge mass, filling level, and/or used engine power.
3. Reduced mill operation time.

Since option 2 and 3 both lead to a reduced amount of produced output per time unit, option 1 is desirable. Option 1 demands the possibility of calculating the strains and stresses in the mill for any process scenario in order to develop and design new mills which will withstand the current variations in loading.

Estimations of the crack propagation speed and the critical crack length, which give the time to failure of the mill, can be obtained by applied fracture mechanics, but require that the stresses in the mill and the material properties should be known. High accuracy in calculations of the stresses is very important in crack propagation calculations, since small errors in the stresses give large errors in the predicted number of cycles before failure or the predicted time to failure.

Models need to be developed which enable precise calculations of the stresses and strains anywhere in the mill and for any process situation. Calculated strains in the mill need to be compared with strain measurements on the real mill in order to verify the models.

Today the loading on the mill is controlled by monitoring the charge mass inside the mill. This is performed by measurements of the pressure on the main supportive bearings of the mill and by calculations based on process data. However, the charge mass is not the only parameter that influences the loading on the mill, since the filling level, charge torque and shape of the upper charge surface also have a major influence (which is described further in Paper A). To avoid
overloading more precisely, it is important that the stresses and strains in the mill can be calculated for the complete loading as a function of the present process situation.

In order to design new mills, to calculate the time to failure and to permit overloading, it must be possible to calculate the stresses, strains, displacements, etc. anywhere in the mill and for any process situation. This is one of the main goals of the work presented in this licentiate thesis.

Calculating the correct strains, stresses, etc. in the mill requires that the correct loading on the mill should be known. Because of the complexity of the load case which the charge inside the mill represents, this is the most challenging part of the present work.

In order to detect cracks at an early stage and to monitor their propagation on the running mills, it is necessary to use some kind of condition monitoring technique. Information about the present machine status and the current crack sizes is useful for adequate maintenance decisions and is important as a complement to the estimation of crack propagation speeds from calculations.

To sum up, the main objectives of the work presented in this thesis are:

1. To develop tools and methods which will enable calculation of the structural strains, stresses, displacements, etc., in the mills in operation at any position in the mills and for any given process situation.
2. To develop tools and methods which will enable calculation of the loading and forces acting on mills in operation for any process situation.
3. To verify the models for calculation of strains and loading with real-life strain measurements on mills in operation and with logged process data.
4. To investigate the loading and strains on mills in operation for different process situations.
5. To find, test and evaluate suitable techniques for crack detection and crack monitoring.

**Scope and limitations**

The work described in this thesis is applicable for any kind of rotating grinding mill. The study is focused on the larger secondary autogenous pebble mills in LKAB dressing plants KA1 and KA2 in Kiruna, with the specific iron ore composition from the local mine. The study can, however, be generalized to apply to any type of tumbling mill, ball mill, semiautogenous mill, etc. and for any ore types.

**Research questions**

To fulfill the purpose and objectives of this research, the following research questions have been formulated:

1. What are the position, shape, volume, mass and torque of the charge inside grinding mills in operation and how can they be calculated?
2. What are the loading and forces acting on grinding mills in operation and how can they be calculated as a function of the process and charge parameters?
3. What are the strains, stresses, displacements, etc. in grinding mills in operation and how can they be calculated as a function of the process and charge parameters?

4. What techniques can be used to detect cracks on grinding mills in operation and how can the crack propagation be monitored without stopping the mill?

Table 1 shows the relationship between the appended papers and the research questions.

<table>
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<th>Research Question</th>
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**Methodology**

As a first step in this work, several condition monitoring and measurement methods were investigated and evaluated based on their ability to detect and monitor fatigue cracks in grinding mills in operation, see Nordström et al. (2009).

After finding the most suitable condition monitoring methods, real-life measurements were carried out on the investigated mills. Wireless strain measurements, thermal measurements and crack propagation measurements were performed on several mills in operation at the LKAB dressing plants. Simultaneously, process data such as the engine power, bearing pressures, charge mass, etc. were measured and collected for the mills. In addition to the field work, this part included learning the different measurement techniques, electrical circuit work to adjust and adapt the equipment for the specific tasks, and laboratory work for testing and tuning.

In parallel with the measurements and data collection, a mathematical model and a computer software were developed for the purpose of calculating the shape, position, mass and torque of the charge inside mills in operation, as well as the loading and forces acting on mills in operation, as a function of process data, strain measurement data and charge parameters. This part included a detailed study of grinding mills and charge behaviour and a substantial amount of mathematical work and programming. Interviews were performed on a regular basis with LKAB staff in order to acquire specific information about the LKAB mills.

In order to calculate the strains and stresses in the mills for any location in the mills and for any process situation, a finite element model of the mills was developed from drawings. This work included studying the mill drawings and specifications and performing a substantial amount of finite element modelling. The finite element modelling included the following parts: creating the best spatial discretization for element representation, geometry simplifications, finding appropriate boundary conditions for mill support, choosing material model and solution type, applying loading, etc. With the developed finite element model the strains in the mill structure were calculated for different load cases obtained by the developed computer software.
The calculated finite element results were compared with the strain measurements and logged process data in order to verify the developed finite element model, load model and computer program. The strain measurement data and logged process data were used both as input for the calculations of the loading and for verification of the calculated results.

Finally, the strain measurements, logged process data, load model, computer software and finite element model were combined together. The result was a system where the strains, stresses, displacements, etc. in the mill can be calculated for any given time and process situation based on the input of the current process and charge parameters. In the final part of the system, the calculated finite element results are verified by strain measurements which give a closed loop from the input of process and charge parameters to the verification of calculated results. The calculation flow is illustrated in Figure 19.

**Evaluation of crack detection and monitoring methods**

In order to find suitable methods for crack detection and monitoring, a large number of condition monitoring methods were investigated and ranked based on pre-defined criteria, see Nordström et al. (2009).

All methods possess their own capabilities and limitations, advantages and drawbacks. The ideal condition monitoring method should be able to detect cracks at an early stage (close to the crack initiation time) and follow the crack propagation with the highest possible accuracy without necessity of stopping the mill. Damage to the mill and/or interference with the production process should be minimized. Moreover, the ideal method must be able to withstand the harsh, wet and dusty environment of dressing plants. Other challenges are the rotation of the mill and the behaviour of the cracks, which sometimes grow in unpredictable ways. The difficulty and complexity of handling and installing the equipment, as well as the cost of the equipment have also been taken into consideration, with easier handling and a lower cost being favoured.

Out of many methods, five were selected for evaluation and the analytical hierarchy process (AHP) was used to find the most suitable one for the application. Based on the results of the AHP, infrared thermography was found to be the most suitable.

**Strain measurements**

In order to understand the loading on the running mills, to verify the calculated strains and to obtain knowledge about the strains in the mill structure, strain measurements were performed on the mills in operation. A total of four measurements were performed, each at different times and with a different placement of the strain gauges. Two measurements were performed on the mill with hydrostatic pressure bearings and the results from these are presented in Paper A and B. Two measurements were performed on the mills with roller bearings, and the results from these measurements are presented in the sections below.
Basics

Strain measurements are performed by strain gauges attached to the object undergoing measurement. Strain gauges usually consist of a grid-shaped metallic resistive foil (3-6 micrometres thick) placed on a base of a thin plastic film (15-16 micrometres thick) laminated with a thin film (Kyowa Electronic Instruments Co., 2012). The strain gauge is tightly bonded to the object subjected to measurement, so that the metallic resistive foil elongates or contracts according to the strain borne by the object. When the metal foil undergoes mechanical elongation or contraction, it undergoes a change in electrical resistance. This change in electrical resistance is used to obtain the strain. The strain is proportional to the electrical resistance change in accordance with a constant of proportionality called the gauge factor, which is specific to each strain gauge type and depends on the type of material in the strain gauge. The change in resistance is very small and impossible to measure with a conventional ohmmeter. The minute resistance changes are therefore measured with a dedicated strain amplifier using an electric circuit called a Wheatstone bridge.

Strain gauges are attached to the object’s surface in the following manner. The surface is first grinded in several steps, first with a grinding machine, then with sandpaper in steps proceeding from rough to fine sandpapering, and finally using wet grinding. Then the object’s surface is thoroughly cleaned with a conditioner and neutralizer. Finally, the gauges are attached to the surface with a special type of glue and a catalyst to speed up the hardening.

Equipment used

Because of the rotating machinery, a wireless strain measurement system was used for all the measurements on the mills. The system used consists of a logger which the strain gauges are connected to and a base station connected to a computer for controlling. Both the logger and the base station have a transmitter and a receiver, and the logger and the computer can therefore communicate by sending and receiving signals to each other. Figure 5 and Figure 6 show how the equipment is typically installed on the mill.

The strain gauges are fixed onto the mill with glue and connected to the logger with cables. The loggers are attached to the mill body with silicon. The logger is powered by a rechargeable battery and controlled wirelessly by the computer. With this setup, measurements can be triggered and stopped remotely at any time. The remote-controlled system is necessary in order to control the measurements without stopping the mill.

The measurements can be performed in two different modes. The first mode requires continuous connection between the logger and the computer during reading, and in this mode the measurements can be seen in real time on the computer screen. The second mode requires no connection between the logger and the computer during reading. In this mode the measurements are triggered and the data are then stored in the logger for later analysis and are not visible in real time. Mode two is the most advantageous for measurements of the mills in operation since the rotation of the mill can create signal losses when the logger rotates away and the signal is covered by the mill body. Therefore, mode two was used for the measurements performed on the mills.
Measurements on a roller bearing mill in operation performed with a synchronizer

Strain measurements were performed on a crack-free roller bearing mill in dressing plant KA1. The mill has a diameter of 5.9 m and a length of 7.7 m. The mill is driven by two pinions, positioned at 3 and 9 o’clock, with one engine each. The total maximum installed power for both engines combined is 3,000 kW. The mill rotates at a constant speed of about 15.1 RPM which is about 86% of the critical mill speed.

Seven strain gauges were attached to the outside of the mill mandrel at six different positions along the mill axis. The gauges were positioned along the same horizontal line, see Figure 5 and Figure 6. The positions of the strain gauges on the mill are marked with yellow dots in Figure 6. The exact positions of the strain gauges were registered in order to allow comparison with numerically calculated strains for the same positions.

Figure 5. a) Strain gauges attached to the mill, b) computer setup during reading

Figure 6. Mill with the measurement setup including a synchronizer, a magnet, loggers, antennas, a computer, and the positions of the strain gauges
For this measurement two loggers were used. Each logger has four channels and each channel has room for one strain gauge, so that each logger holds four strain gauges. An L-shaped strain sensor consisting of two strain gauges perpendicular to each other was attached at position 3 in Figure 6, with one strain gauge oriented circumferentially (Cir) and the other longitudinally (Long). At the other positions in Figure 6, only one strain gauge was used, each oriented circumferentially. It must be noted that strain gauges measure strains in the directions in which they are oriented.

In order to synchronize the measured strains with the circumferential position on the mill mandrel, a magnetic synchronizer and a magnet were used. The synchronizer was connected to an empty channel on one of the loggers. The synchronizer changes the electrical resistance in the bridge channel when passing a magnetic field. A strong magnet was placed on the foundation near the mill at the 9 o’clock position with respect to the mill seen from the inlet end, see Figure 6 and Figure 7. When the mill rotates, the synchronizer passes the magnet which gives a strong peak voltage signal. The peak signal is visible on the same graph as that showing the measured strains of the strain gauges connected to the same logger. The measured strains can then be related to the known 9 o’clock position on the mill and the exact angular positions of the strains in the mill can be known. With this setup the positions of the strains on the mandrel are known both longitudinally and circumferentially. Figure 7 shows the logger attached to the mill mandrel, the synchronizer attached to the mill flange and the magnet placed on the top of a metal stick.

To avoid unnecessary production stoppages, the equipment was attached to the mill during a maintenance stoppage. The measurements were then performed after the mill had been restarted and the production had stabilized.

The strains were measured with the synchronizer attached with a frequency of 512 Hz during several mill rotations. The results from one measurement are shown in Figure 8, where the strains are plotted against time. As can be seen in Figure 8, the strain pattern is similar and repetitive, with almost the same strain range in each cycle. Each repetitive pattern corresponds to one 360° rotation of the mill. The dotted vertical lines in Figure 8c and Figure 8d are the voltage pulses from the synchronizer.
The raw measurement data give quite rough data plots, as can be seen in Figure 8a and Figure 8c, which are difficult to analyse. In order to smoothen out the plots, signal processing was used and the smoothened plots are visible in Figure 8b and Figure 8d. The signal processing code used for this purpose uses gliding mean values to smoothen out the curves. All the strain values $ε(i)$ (where $i = 1, 2, 3, 4,..., n$) in the measured strain data series are replaced by the mean value $ε_m(i)$ of the values in a window between $i - k$ and $i + k$. The formula for $ε_m(i)$ is given as:

$$ε_m(i) = \frac{1}{2 \cdot k + 1} \sum_{j=-k}^{k} ε(i + j) \tag{1}$$

The $k$-value gives the size of the window and the grade of smoothness of the curves, with a higher $k$ giving smoother curves and vice versa. A $k$-value of 100 was used to process the present strain data. $k = 0$ gives untreated data curves.

Table 2 shows the average measured strain range for one mill rotation at each mill location and for each direction, obtained from the raw non-signal-processed data. The table shows that the circumferential strain range is higher near the inlet end and decreases towards the outlet end. This could indicate that the charge mass is higher near the inlet end and decreases towards the outlet end.

The strain gauges were fixed and calibrated on the mill on an occasion when the mill was idle and still loaded, and therefore strains already existed in the structure. Consequently, the obtained results show only the variation of the strains during each rotation and not the absolute values of the strains in the structure.
Figure 8. Measured strains on the mill mandrel during several mill rotations, a) and c): raw data, b) and d): signal-processed data

Table 2. Average measured strain range for one mill rotation

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<th>Position</th>
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<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
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<tr>
<td>Measurement direction</td>
<td>Cir</td>
<td>Cir</td>
<td>Cir</td>
<td>Long</td>
<td>Cir</td>
<td>Cir</td>
<td>Cir</td>
</tr>
<tr>
<td>Strain range [μm/mm]</td>
<td>8.54</td>
<td>10.02</td>
<td>9.39</td>
<td>17.14</td>
<td>12.72</td>
<td>11.43</td>
<td>17.77</td>
</tr>
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Seen over a longer period of time, the mill is subjected to fatigue loading of variable amplitude because of the characteristic changes in the process and charge parameters, e.g. the engine power, the charge mass, the relative filling level inside the mill, the shape of the upper charge surface, etc. However, seen over a short period, the measurements indicate that the mill is subjected to fatigue loading of almost constant amplitude.

The measured results shown in Figure 8 give information about the present strains in the structure which can be used to permit overloading, indicate cracks and verify the finite-element-calculated results. The measured strains can also be used for estimations of the filling level, the shape of the upper charge surface and the position of the charge circumferentially inside the mill.

In Figure 9 the measured signal-processed strains at position 4 and 5 for a single mill rotation are shown. With the help of the synchronizer, the strains are plotted against the angular position on the mill mandrel. Figure 10 shows the angular starting point (Φ=0°) and direction used for the plots.

The shape of the strain curves is related to the shape, position, mass and volume of the charge inside the mill. The synchronizer gives the exact angular positions of the strains in the mandrel, see Figure 9. The global minimum, at location 1 in the plot (Φ=38.2°), is equivalent to the toe position of the charge, and the local minimum, at position 2 in the plot (Φ=191.9°), is related to the shoulder position of the charge. Based on this, the position of the toe and shoulder can be obtained, which gives the angular position of the charge, which is denoted by the parameter α, see Figure 17a.

The toe is the lower part of the upper charge surface touching the linings and the shoulder is the higher part of the upper charge surface touching the linings, see Figure 10a.
As can be seen in Figure 9, the global maximum is located at the bottom of the mill where most strains occur (around $\Phi = 270^\circ$); at this position the bending is highest and the effect of the self-weight of the mill is largest.

The angular distance between position 1 and 2 in Figure 9 is directly related to the shape of the upper charge surface and the filling level inside the mill. The shape of the upper charge surface is given by the parameter $R$, which is the circular radius of the charge surface, see Figure 17a. A smaller distance between position 1 and 2, for a constant $R$, means a higher filling level and a larger distance means a lower filling level. In the same way, a smaller distance between position 1 and 2 indicates a smaller $R$ value for a constant filling level and an increasing distance indicates a larger $R$ value. To provide an example to illustrate the above-described properties, the shape and position of the charge for the time period during the performed measurements have been calculated using the developed MLC software and equations described in the appended Paper A and B.

During the measurements a total charge mass of 254.6 tons and a total engine power of 2,777 kW were obtained from the logged data. The mill rotation speed was calculated to be 15.19 RPM from the time per revolution between measured sync pulses.

The relative filling level inside the mill is typically in the range of 35-40% for normal production. Filling levels over 45% lead to back-flow. The logged charge mass and a filling level of 37.5% give a calculated charge density of 3,594 kg/m$^3$, which is considered to be within the normal production limits. By using the above known parameters and $R = \text{Inf}$ (i.e. a straight upper charge surface), the charge profile, as seen from the outlet end, according to Figure 10a was obtained from calculations. The inner circle represents the outlet/inlet hole of the mill.

During production the charge surface is concave and the correct $R$ value is obtained by adjusting $R$ so that the angular positions of the toe and shoulder agree with the corresponding measured ones. By doing this, $R = 3,150.4$ mm gives the charge profile seen in Figure 10b, where the positions of the toe and shoulder are in agreement with the measured values shown in Figure 9. Now all the geometrical charge features are known, e.g. the upper charge surface shape, the charge volume and the position of the charge circumferentially inside the mill, which gives great advantages when calculating the loading from the charge for structural analysis (which is described later, and in Paper A and B). The $R$-value, or charge upper surface shape, was here obtained from the strain measurement data by using the synchronizer.

The fluctuating strains in the mandrel between position 1 and 2 in Figure 9 were caused by mechanical waves in the mill structure introduced by the movement of the charge. This phenomena is described by Jonsen et al. (2011).
Measurements on a roller bearing mill in operation containing a large crack

In one of the roller bearing mills, of the same type as that described above, a large circumferential crack of a visible outer length of 1.5 m was found in the mandrel near the fillet weld and the flange at the inlet end, see Figure 1. The position of the crack on the mill is shown in Figure 6. The crack was repaired by welding triangular plates over the crack, see Figure 3. In order to investigate how the strains in the mill were affected by the welded crack, strain measurements were performed on the cracked mill in operation. A three-gauge strain sensor was placed on the mill mandrel, near the crack, at a distance of 0.25 m from the inlet flange (close to position 6 in Figure 6), see Figure 11. The attached sensor consists of three strain gauges, two oriented perpendicular to each other and one oriented at an angle of 45° to the other two.

The sensor was placed with the perpendicular gauges directed circumferentially and longitudinally, respectively. The strains were measured during several mill rotations and the results are shown in Figure 12. Table 3 shows the average measured strain ranges in each direction.
During these measurements the charge mass was logged as 224.6 tons, and the mill rotation speed was calculated as 15.15 RPM from the time between the local maximums of the strain cycles in Figure 12. The total engine power for both engines was logged as 2,126 kW.

Since the position of the strain gauges for this measurement is about the same as that for the measurements on the mill of the same type without a crack described earlier, these data can be compared directly to each other. One can see that the circumferential strain range at this location on the cracked mill is significantly higher than that for the non-cracked mill. It should be noted that the logged charge mass and engine power were lower during the measurements on the cracked mill. The measured strain range is obviously higher on the crack mill even though the loading is less. One conclusion is that a crack, even though it is welded, increases the measured strain range in the mill significantly during operation. This means that strain gauges can be used as a condition monitoring tool for crack detection on mills in operation. When a crack appears, the strain range increases above the usual levels and indicates the crack.

![Attached crack gauge and strain gauges near the welded crack at the inlet end](image1)

![Measured strains near a welded crack for several mill rotations](image2)

<table>
<thead>
<tr>
<th>Table 3. Average measured strain range for one mill rotation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Position</td>
</tr>
<tr>
<td>Direction</td>
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<tr>
<td>Strain range [μ mm/mm]</td>
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</tbody>
</table>
Online crack propagation measurements

Online crack propagation measurements were performed on the welded fatigue crack in the mill described in the above section. The crack propagation was measured with a crack sensor attached to the crack tip of one of the smaller cracks growing out from the larger one. The measurements were performed with the same wireless system as that used for the strain gauge measurements, with the possibility of triggering and stopping measurements remotely.

A crack sensor is constructed in the same way as a strain gauge with a matrix of thin electric conductive wires inside a thin polymer film. The sensor is about 8-10 times larger than a normal strain gauge and the metal matrix consists of several parallel wires, see Figure 13a. The sensor is attached in front of the crack tip with glue. The attachment process is the same as that for a strain gauge. As the crack propagates through the sensor, the wires break one after another. As the wires break, the resistance increases in the sensor and, according to Ohm’s law, the voltage increases in the circuit. The voltage level is measured by the logger and the crack propagation can then be followed as a function of the increasing voltage level. Figure 13a illustrates how a crack grows into the sensor matrix and how wires break as the crack propagates. An example curve for the crack growth increment ($\Delta a$) and voltage ($U$) as a function of time is provided in Figure 13b. Since the crack growth is indicated by the breakage of wires, the propagation is discrete and step-wise, as shown by the black curve in Figure 13b.

![Figure 13. a) Crack sensor with a growing crack, b) crack length/voltage increase versus time](image)

The increase in voltage per wire break is unique for each circuit and needs to be tested in the laboratory before measurements. For the specific measurement setup used for the mill in question, a voltage increase of 10 volts per wire break was noted during laboratory testing. The sensor consists of 26 wires and the breakage of one wire is equivalent to about 1 mm of crack growth.

Usually a long logging time is required for this type of measurement and, in order to save battery power, the sampling frequency is usually kept low, often less than one reading/hour. If a high crack propagation speed is expected, the sampling frequency must be set higher to capture the rapid crack growth.

During the measurements performed on the mill, only two readings could be taken, one at the installation of the crack sensor and one at the replacement of the cracked part and the removal of the equipment three months later. The voltage increase during this time period was noted as 23 volts which represents a crack growth of about 2 mm.
Thermography measurements

An infrared (IR) thermal camera and an IR scanner were used as methods to find cracks and other material damage in several grinding mills in operation. This is described in detail in Paper C.

The temperature is higher inside the mill than outside and cracks can therefore be found through the increasing temperature arising from heat passing out through the crack openings.

The IR camera used had a sampling rate of 30 Hz, a temperature sensitivity of 0.06°C and a temperature range from -40°C to 500°C. The scanner used had a sampling rate of 100 Hz, a temperature sensitivity of 0.08°C and a temperature range from -20°C to 900°C.

The camera had the advantage of having a high resolution, but the disadvantage of having a low sampling rate, making it difficult to capture pictures of the mill in operation rapidly. The scanner had a low resolution, but a high sampling rate, which enabled the fast capturing of pictures of the running mill. However, the low resolution required the scanner to be placed very close to the mill, often at a distance less than 0.5 m for the typical rotation speed and sizes of the investigated mills, which made it less practical.

The camera was found to be most useful for the application in question because of its higher resolution, and it seems that a high resolution is more important than a high sampling rate for this application.

Numerical simulations and modelling

Strain measurements cannot be used to obtain strains at any location on the mill since strain gauges cannot be placed near corners, bolt holes or other stress raisers. In order to obtain the strains and stresses in the mill at any location, and for any process situation, structural analysis with numerical methods is necessary. The strains, stresses, displacements, etc. in the mill were therefore calculated using the finite element method. This work is described in detail in Paper A and B. Below a summary of this work is presented, with a focus on the methodology used.

The finite element method

The finite element method (FEM) is a numerical approximate analysis method where searched fields are approximated by functions defined over sub-areas based on values in a number of nodal points on the boundaries between them. The most common form is based on displacement approaches. The introduced quantities are determined by conditions of the solution by some form of weak formulation of the problem, for the displacement-based approach and for approaches based on the principle of virtual work, the potential energy minimum or the Galerkins method (Ottosen and Petersson, 1992; Zienkiewicz and Taylor, 2000).
Structural analysis

A structural analysis with the finite element method was performed for a mill supported by hydrostatic bearings and placed in dressing plant KA2.

The mill has a diameter of 6.5 m and a length of 8.5 m, and is driven by one pinion, positioned at 20° under the horizontal line. The maximum installed engine power is 4,500 kW. The mill rotates at a constant speed of 12.75 rpm (revolutions per minute), which is about 75% of the critical mill speed.

Figure 14. Finite element model of the mill, a) finite element mesh and applied boundary conditions, b) cross-section in the Y-Z plane, with visible meshed linings and discharger

Figure 15. Calculated strains in the mill, a) circumferential strains, b) longitudinal strains
For the structural analysis the following steps were performed:

1. Literature survey and detailed investigation of the function of the mill and the grinding process.
2. Detailed study of the geometry and material properties of the mill based on drawings, instruction books, manual inspections, interviews with mill operators, etc.
3. Investigation of the bearing supports of the mill to be able to apply the most appropriate boundary conditions for the finite element model.
4. Detailed investigation of the charge behaviour inside the mill and the loading on the mill in operation; necessary in order to apply the correct loading in the finite element model.
5. Development of a mathematical model for calculation of various charge features, e.g. the charge torque, the angular position of the charge circumferentially, the shape of the upper charge surface, the charge mass, charge density, filling level, etc, for the mill in operation. All of these details are necessary in order to calculate the loads from the charge.
6. Development of a computer software named the “Mill Load Calculator (MLC)” for calculation of the loading and forces acting on the mill in operation for input of the process and charge parameters. Figure 20 shows the user interface of this software.
7. Creation of a computerized model of the mill based on drawings, together with an applied material model in accordance with the material properties of the mill.
8. Simplification of the detailed model in order to facilitate discretization into finite elements and to reduce the computational time. The simplified model included only those details which were deemed relevant to the structural stiffness; e.g. bolt holes, small corner radii and similar details were neglected.
9. Discretization of the simplified model into a finite element mesh with 3D brick elements for the mill body, a 0D point mass for the trommel sieve and 2D rigid elements for the connection between the trommel mass and the mill body, see Figure 14a.
10. Application of boundary conditions in order to mimic the bearing supports of the real mill. The mill model was locked in the radial direction only at the contact surfaces for the six main bearings and locked in the axial direction only at the contact surface for the axial bearing. One node was locked in the rotational direction only to prevent singularity of the global element stiffness matrix during solving.
11. Application of loading in terms of gravity and forces from the charge and pinion for quasi-static loading, see Figure 21 to Figure 24.

12. A convergence study of the finite element mesh was performed, especially at the location of the strain gauges, where the highest accuracy is desired. The finite element mesh was refined in several steps with the purpose of finding the largest possible element size without decreased accuracy.

13. Test of different element types, such as 8 node linear approximation and 20 node quadratic approximation, with the purpose of finding the most appropriate element type. Finally, 8 node elements were used for the model since the results for the two element types were equal and the 8 node elements are computationally less demanding than the 20 node elements.

14. Meshing and incorporation of the linings, discharger and parts of the ore transition system into the existing model and finding the appropriate stiffnesses of these parts, see Figure 14b.

15. Solving the numerical model, with small deformation theory, for different load cases. Figure 15 shows examples of calculated circumferential and longitudinal strains in the mill. Figure 16a shows the Von Mises stresses calculated in the mill for a load case, and the picture reveals the most loaded half of the mill with the linings and discharger hidden, showing only the stresses in the steel shell.

16. Comparison of the calculated results with the strain measurements and the logged bearing reaction forces for validation of the model and loading. Figure 16b shows an example of a comparison between the calculated and the measured strains for one mill rotation at the axial position of the applied strain gauges.

**Charge and loading analysis**

For the structural analysis, the loads acting on the running mill must be known. The loading on the mill in operation comes from the charge inside the mill, the reaction forces at the bearing supports, the drive wheel pinion and the self-weight of the mill and attached components.

Since the production process is constantly changing the forces from the charge, pinion and bearing supports vary with time. As a first step it was necessary to find out how the forces from the charge and pinion are related to the process and charge parameters.

In order to determine the loads due to the charge, the shape, volume and position of the charge inside the running mill must be known, as well as the total charge mass and charge torque.

In operation the upper surface of the axial cross-section of the charge is typically concave in shape, see Figure 17a. With regard to volume, the charge distribution is fairly uniform in the axial mill direction, but the charge density varies and decreases often towards the outlet end, see Figure 17b.

During operation the motor power rotates the charge to one side, see Figure 17a. The position of the charge circumferentially (α) is obtained from the power equilibrium of the mill and motor. The displacement of the charge, from the zero torque position, is a function of the motor power
(\(P_{\text{motor}}\)), the mill rotation speed (\(\omega\)), the total equivalent charge mass (\(m\)), the filling level (\(V\)) inside the mill, the radius of the upper charge surface (\(R\)), the inner mill radius (\(r\)), the factor for the loss of power in each gear transmission through the drive-train (\(\eta\)) and the total number of gear transmissions (\(n\)).

The torque of the mill can be written as:

\[
M_{\text{mill}} = \frac{P_{\text{motor}} \cdot \eta}{\omega}
\]

and the torque of the charge can be written as:

\[
M_{\text{charge}} = d(r, R, V, \alpha) \cdot m \cdot g
\]

\(d\) is the horizontal distance between the mill centre and the centre of gravity (CG) of the charge. \(g\) is the acceleration due to gravity. \(d\) is a function of \(r\), \(R\), \(V\) and \(\alpha\), where \(\alpha\) is the angular displacement of the charge’s CG counter-clockwise with respect to a vertical line through the centre of the mill.

In operation, the torque of the charge equals the mill torque:

\[
M_{\text{charge}} = M_{\text{mill}}
\]

By combining equation (2), (3) and (4) the position of the charge circumferentially can be written as a function of \(r\), \(R\), \(V\), \(m\), \(g\), \(P_{\text{motor}}\), \(\eta\), \(n\) and \(\omega\):

\[
\alpha = (r, R, V, m, g, P_{\text{motor}}, \eta, n, \omega)
\]

For each specific mill, \(r\), \(g\), \(n\), and \(\eta\) are constants, while \(R\), \(V\), \(m\), \(P_{\text{motor}}\) and \(\omega\) vary with production and depend on the specific time and process situation. For the full expressions of equation (3) and (5), see the appended Paper A and B.

The charge mass is obtained from the logged bearing pressures by subtraction of the empty mill and attached components from the total mass logged on the bearings. The total density of the charge (\(\rho\)) can then be calculated from \(V\) and \(m\) with the following formula:

\[
\rho = \frac{m}{V \cdot L \cdot \pi \cdot r^2}
\]

where \(L\) is the inner length of the mill.
In order to calculate the loading on the mill, the charge is assumed to act like a fluid exerting pressure and friction on the mill. Based on this assumption, the loads on the mill are the following: normal forces \((F_{\text{normal}})\) and friction forces \((F_{\text{friction}})\) from the charge acting on the mandrel and the heads; tangential forces \((F_{\text{wheel}})\), normal forces \((F_{\text{wheel normal}})\) and axial forces \((F_{\text{wheel axial}})\) from the pinion acting on the drive wheel; gravity forces corresponding to the weight of the mill itself and the attached components; and radial and axial reaction forces at the bearing supports \((F_{\text{bearing}}\) and \(F_{\text{bearing axial}})\), the latter at the outlet end only.

Figure 6 shows simplified free body diagrams of the forces acting on the mill in operation. The green arrows represent the forces acting on the mill heads and the red arrows the forces acting on the mandrel. Friction forces between the mill and the hydrostatic bearings are neglected.

During operation, in addition to equation (4), the following torque equilibrium condition exists for the mill:
$M_{\text{charge}} = M_{\text{friction}} = M_{\text{wheel}}$  \hspace{1cm} (7)

where $M_{\text{friction}}$ is the total torque from the friction forces acting on the mill from the charge, $M_{\text{wheel}}$ is the total torque from the tangential forces acting on the drive wheel, and $M_{\text{charge}}$ is given by equation (3) and (4).

**Calculation of the loading on the mill in operation**

This section describes the calculation of the forces acting on the mill from the charge and pinion as a function of the process, mill and charge parameters.

**Known inputs and initial calculations**

To be able to calculate the forces from the charge and the pinion, a number of parameters related to the charge, process and mill geometry must first be known. The necessary steps to obtain these parameters are described below:

1. The necessary geometrical constants, according to the mill design, are obtained from drawings. These are the inner mill length ($L$), the inner mill radius ($r$), the factor for the loss of power in each gear transmission through the drive-train ($\eta$) and the total number of gear transmissions ($n$), the gear data for the drive wheel and pinion, the masses of the empty mill and attached components such as the linings, discharger and trommel, the thicknesses of the linings and discharger, and the radius of the inlet and outlet hole of the mill.

2. The total equivalent charge mass ($m$) varies with production and can be obtained from the logged bearing pressures by subtraction of the empty mill and attached components from the total mass logged on the bearings, see Paper A. In the case where the bearing pressures are not logged, $m$ can be obtained from a special type of mathematical estimation based on other process data (not described here).

3. The filling level inside the mill ($V$) varies with production, but is often, according to past experience of the process, within a narrow interval and is typically in the range of 30-45%. This parameter can be obtained by manually measuring the vertical charge depth with a measuring stick during a mill stoppage. The parameter can also be obtained using various online measurement techniques such as vibration or strain gauges mounted in liner lifters. During this research it has been found that $V$ can be estimated from the measured strains on the mill mandrel (described under the strain measurements section). $V$ can also be estimated by comparing the charge density ($\rho$), calculated from $V$ and $m$, and the theoretical value of $\rho$ based on the composition of the charge and obtained from knowledge of the mineral type, crushing grade, particle size distribution and amount of water in the charge (described in more detail in Paper A and B).

4. The engine power used ($P_{\text{motor}}$) varies with production and is frequently logged.

5. The mill rotation speed ($\omega$) is about constant during production. $\omega$ can be obtained directly from strain measurement data, from the logged time between two strain cycles or the logged time between two sync pulses, which gives the time per revolution of the mill (described in the above strain measurements section).
6. The concavity radius of the upper charge surface (\(R\)) is an important parameter that denotes the shape of the charge upper surface. \(R\) can be obtained by comparing the measured strains with the finite-element-calculated strains for different values of this parameter until the best possible agreement is achieved, which then gives \(R\) (described in Paper A). Another way to obtain this parameter is to use strain measurements and a synchronizer on the mill mandrel during production (described in the above strain measurements section).

7. The acceleration due to gravity (\(g\)) is related to the location of the mill on the earth and can be obtained from literature.

At this stage, \(m, V, P_{\text{motor}}, \omega, g, r, R, L, n, \eta\), the gear data, the masses of the empty mill and attached components, and other geometrical dimensions are known, which makes it possible to go further to the next step.

**Intermediate calculations**

At this step, the angular position of the charge, the charge torque and the charge density can be calculated by the following sub-steps:

1. The angular displacement of the charge’s CG with respect to a vertical line through the centre of the mill (\(\theta\)) is calculated from \(m, V, P_{\text{motor}}, \omega, r, R, g, n, \eta\) with equation (5) (for more detailed explanations see Paper A and B).
2. The charge torque (\(M_{\text{charge}}\)) is calculated from \(P_{\text{motor}}, \omega, n, \eta\) and equation (2) and (4) (see also Paper A and Paper B).
3. The charge density (\(\rho\)) is calculated from \(V, L, m, r\), and equation (6).

Now all the parameters are known which are necessary to calculate the forces acting on the mill from the charge and the pinion.

**Final calculations of the forces acting on the mill from the charge and the pinion**

**Forces from the charge:**

1. From \(V, R \) and \(a\), the volume, shape and position of the charge are known. Further, \(g, p, L, r\), the radius of the inlet and outlet hole, and the thicknesses of the linings and discharger are known. On the basis of this, the normal forces acting on the mill from the charge are calculated with equation (9) and a special type of algorithm in the MLC software explained under the section below entitled “The charge load calculation algorithm”. A detailed distribution of hundreds of thousands of normal forces, each with its own direction, magnitude and position, is calculated at this stage. The information about the forces is stored in platform-independent files for later application in the finite element model. The total number of forces can be set to any value and is only limited by the computer capacity.
2. The total torque from the friction forces acting on the mill (\(M_{\text{friction}}\)) is calculated from \(M_{\text{charge}} \) and equation (7).
3. Using \(M_{\text{friction}}, V, R \) and \(a, g, p, L, r\), the radius of the inlet and outlet hole, and the thicknesses of the linings and discharger, the coefficient of friction between the charge
and the mill ($\mu$) is calculated with equation (13) and a special algorithm in the MLC software.

4. From the calculated normal forces and by using $\mu$, $V$, $R$, $\alpha$, $L$, $r$, the radius of the inlet and outlet hole, and the thicknesses of the linings and discharger, the friction forces acting on the mill from the charge are calculated with equation (10) and a special algorithm in the MLC software. As was performed for the normal forces, a detailed distribution of hundreds of thousands of friction forces, each with its own direction, magnitude and position, is calculated. The information about the forces is stored in platform-independent files for later application in the finite element model. The total number of forces can be set to any value and is only limited by the computer capacity.

Forces from the pinion:

1. As a first step, the total torque from the tangential forces acting on the drive wheel ($M_{\text{wheel}}$) is calculated with $M_{\text{charge}}$ and equation (7).
2. The tangential, normal and axial forces acting on the drive wheel from the pinion are then calculated from $M_{\text{wheel}}$ and the gear data for the drive wheel and pinion, e.g. the addendum modification factors, number of teeth, pressure angle, helix angle and real module. This is described more detailed in Paper A.
3. A detailed distribution of the forces, each with its own direction, magnitude and position, is calculated with a special algorithm in the MLC program and stored in files for later application in the finite element model.
4. The wheel forces are distributed linearly on the drive wheel over an angular range of 20° and symmetrically around the contact point with the pinion, with the maximum at $\phi=340°$ and zero again at $\phi=330°$ and $\phi=350°$, see Figure 20 and Figure 24.

The MLC software

In order to automate the calculation of the charge distribution and the loads acting on the mill for different charge and process parameters, a computer software called the Mill Load Calculator (MLC) was developed.

The software calculates the charge distribution, $\alpha$, $m$, $M_{\text{charge}}$, the force distributions, $F_{\text{normal}}$, $F_{\text{friction}}$, and the forces $F_{\text{wheel}}$, $F_{\text{wheel normal}}$ and $F_{\text{wheel axial}}$ from the input of the bearing pressures, $\alpha$, $V$, $P_{\text{motor}}$ and $R$. The program also delivers the charge density, $\rho$, the present trommel mass, $m_{\text{trommel}}$, the specific pebble and pulp density, $\rho_{\text{pebbles}}$ and $\rho_{\text{pulp}}$, and the coefficient of friction between the charge and the mill, $\mu$.

The bearing pressures for the main bearings are denoted, according to the standard of the mining company, by $P_{112}$, $P_{113}$, $P_{114}$, $P_{115}$, $P_{116}$ and $P_{117}$, see Figure 20.

In addition to the above-mentioned outputs, the software also visualizes the charge distribution, along with the position of the charge’s CG, in 2D and 3D, together with the distributions of the forces acting on the mill, in the graphical user interface, see Figure 20.
The axial charge mass distribution is visualized with two bars in Figure 20, each bar indicating how much of the total charge mass is portioned at each mill end. The charge surface of the 3D plot has a faded colour, and there is a darker colour where the charge density is higher. This indicates further the axial charge density distribution inside the mill.

The calculated loads and their distributions are exported to the finite element software. The computational mill model is updated and a solution is calculated for the current load case. The data transfer is platform-independent. Figure 19 shows a block diagram of the calculation work flow including the MLC software. Based on the MLC software data, the finite element model yields the stresses, strains, reaction forces, displacements, etc. in the mill for any given input of process and charge parameters.

Figure 21 to Figure 23 show the finite element model with examples of force distributions and applied forces from the charge for different process and charge parameters. The mesh grid is hidden here in order to make the pictures more clear. Figure 24 shows the typical distribution of the forces from the pinion on the drive wheel.

![Figure 19. Block diagram of the calculation work flow](image-url)
Figure 20. Graphical user interface for the MLC software

Figure 21. $V=40\%$, $P_{motor}=0$ kW, $\omega=0$ rpm, $m=372.1$ tons, $\rho=3570$ kg/m$^3$, $M_{charge}=0$ kNm, $R=Inf$ mm, $\alpha=0^\circ$ a) charge profile with the CG marked, seen from the outlet end, b) distribution of normal forces from the charge, red=max and blue=min, c) applied normal forces from the charge
Figure 22. $V=30\%$, $P_{\text{motor}}=4500\ kW$, $\omega=12.75\ rpm$, $m=372.1\ tons$, $\rho=4760\ kg/m^3$, $M_{\text{charge}}=3172\ kNm$, $R=3573.9\ mm$, $\alpha=28.25\degree$; a) charge profile with the CG marked seen from the outlet end, b) distribution of normal and friction forces from the charge, red=max and blue=min, c) applied normal forces from the charge

Figure 23. $V=42.5\%$, $P_{\text{motor}}=3000\ kW$, $\omega=12.75\ rpm$, $m=372.1\ tons$, $\rho=3360\ kg/m^3$, $M_{\text{charge}}=2115\ kNm$, $R=3772.4\ mm$, $\alpha=23.90\degree$; a) charge profile with the CG marked seen from the outlet end, b) distribution of normal and friction forces from the charge, red=max and blue=min, c) applied normal forces from the charge

Figure 24. $P_{\text{motor}}=3000\ kW$, $\omega=12.75\ rpm$, $M_{\text{ax}}=2115\ kNm$, a) distribution of normal, tangential and axial forces from the pinion, red=max and blue=min, b) applied normal forces from the pinion
The charge load calculation algorithm

Since no commercial FEM software found included any specific routine for calculating and distributing the loads from the charge in a mill in operation, an algorithm for this purpose was developed by the author and embedded into the MLC software. The basis for this algorithm is explained below.

The form, volume and orientation of the charge inside the running mill are given by the parameters $R$, $V$ and $\alpha$. The area where the charge forces act is denoted by $S_{\text{mandrel}}$ for the mill mandrel and $S_{\text{head}}$ for the mill heads. $S_{\text{head}}$ is the total mill head area in contact with the charge (coloured light blue in Figure 25). $S_{\text{mandrel}}$ is the total mill mandrel area in contact with the charge between $\Phi = 180^\circ$ and $\Phi = 360^\circ$, see Figure 25.

In order to obtain the forces from the charge, the surfaces $S_{\text{mandrel}}$ and $S_{\text{head}}$ are divided into a large number, $m$ and $n$, respectively, of discrete area elements, each with an area denoted by $A_{\text{mandrel}}$ and $A_{\text{head}}$, respectively. The position of each area element is calculated. The pressure from the charge acting on an area element is given by the hydrostatic law:

$$p_{\text{element}} = \rho \cdot g \cdot h$$  \hspace{1cm} (8)

where $h$ is the vertical charge height above the element. The normal force acting on the element can then be written as:

$$F_{\text{normal, element}} = p_{\text{element}} \cdot A = \rho \cdot g \cdot h \cdot A$$  \hspace{1cm} (9)
where \( A \) is the area of the element. Since \( h \) is calculated vertically from a surface element and upwards, no forces are acting on the mandrel between \( \Phi = 0^\circ \) and \( \Phi = 180^\circ \), see Figure 25.

The friction force acting on an area element is obtained by multiplying the normal force acting on the element by the coefficient of friction \( \mu \) between the charge and the mill:

\[
F_{\text{friction, element}} = \mu \cdot F_{\text{normal, element}}
\]  
(10)

As can be seen in equation (9), the magnitude of the normal and the friction force at a given position is directly proportional to \( h \) at the position in question. The torque from friction on an area element is given as:

\[
M_{\text{element}} = F_{\text{friction, element}} \cdot r_{\text{element}}
\]  
(11)

where \( r_{\text{element}} \) is the radius to the element. The total torque from the friction forces acting on the mill is the sum of the products of the friction force acting on an area element and the corresponding radial distance from the centre of the mill:

\[
M_{\text{friction}} = \sum M_{\text{element}} = \mu \cdot \rho \cdot g \cdot \left( A_{\text{mandrel}} \cdot r \cdot \sum_{i=1}^{n} h_i + 2 \cdot A_{\text{head}} \cdot \sum_{j=1}^{n} h_j \cdot r_j \right)
\]  
(12)

In this equation, \( h_i \) is the vertical charge height above the area element at position \( i \) on \( S_{\text{mandrel}} \) and \( h_j \) is the vertical charge height above the area element at position \( j \) on \( S_{\text{head}} \). \( r_j \) is the radius to the area element at position \( j \) on \( S_{\text{head}} \), see Figure 25. Rewriting (12) gives the coefficient of friction between the charge and the mill as:

\[
\mu = \frac{M_{\text{friction}}}{\rho \cdot g \cdot \left( A_{\text{mandrel}} \cdot r \cdot \sum_{i=1}^{n} h_i + 2 \cdot A_{\text{head}} \cdot \sum_{j=1}^{n} h_j \cdot r_j \right)}
\]  
(13)

where \( M_{\text{friction}} \) is known from equation (7). Inserting equation (12) into (10) gives the friction force at any location.

From the algorithm and the equations described above the normal and friction force distributions from the charge are calculated with hundreds of thousands of forces. The total number of forces can be set to any value and is only limited by the computer capacity.

The distributions of the area element’s normal and friction forces are described by the coordinates for the discrete positions of the forces over \( S_{\text{mandrel}} \) and \( S_{\text{head}} \). The force distribution between the discrete points is interpolated in the FEM-software.
The information about the magnitude and distribution of the forces is stored in platform-independent files which are used for the communication between the MLC software and any finite element software.

**Discussion of results and conclusions**

In this chapter the research work leading to this thesis and the research results from this research are discussed and conclusions are presented with answers to the research questions.

**Crack detection and monitoring techniques**

It has been found that strain measurements during mill operation with strain gauges attached to the mill mandrel are a useful method for detection of larger circumferential cracks near the flanges in the mill. A crack can be detected through the increasing measured strain range due to the crack. The strain range is expected to increase with the crack size, which also enables crack propagation monitoring. The strain gauges should be attached close to the expected position of the cracks. Since the cracks often appear at the same locations, in fillet welds near flanges at the inlet and outlet ends, the correct installation of the strain gauges is predictable. In order to apply this method for online monitoring of cracks, a certain type of self-charging mechanism can be installed on the mill which re-charges the batteries in the logger while the mill is running. One limitation of this method is that the exact sizes of the cracks are difficult to ascertain but as means of signalling alarms for cracks, the method works well.

Infrared thermography is most useful for the detection of larger cracks. Also here the exact crack size is difficult to monitor, but the method gives a quick rough estimation of the present health status of the machine. As the thermal sensitivity of cameras increases, it will be possible to obtain a more detailed thermal field with finer gradations, which will enable more precise monitoring of the machine status. The method is easy and fast to use and the cost of the equipment is low, which makes the method a good complement to strain measurements for the occasional quick scanning of mills while they are in operation, in between maintenance stoppages.

Crack sensors have been found to be a useful method to monitor the propagation of the smaller cracks located at the corners of manholes. The sensors give very precise measurements of the crack growth. The method can be used for crack detection as well as for crack monitoring. If the sensors are attached to the places where cracks are expected, the cracks are then detected when they grow into the sensor matrix. For online monitoring with crack sensors, as was possible for strain measurements, a battery-self-recharging mechanism can be installed on the mill.
To sum up the findings of this part:

- For the detection and monitoring of larger circumferential cracks near flanges, fillet welds, etc., strain gauges and the thermal camera are the most advantageous methods to use.
- For the detection and monitoring of smaller cracks at the corners of manholes, crack sensors are the wisest choice.

Hopefully these methods will meet most crack condition monitoring needs. All the methods have been tested and evaluated on real mills in operation. This part gives answers to research question 4.

**Strain measurements**

Strain measurements have been found useful, not only for crack detection and monitoring, but also for the purpose of validating finite-element-calculated results and estimating charge features such as the filling level, the radius of the upper charge surface, and the position of the charge circumferentially inside the mill in operation. Strain measurements are certainly effective when using a synchronizer (as described in the section “Measurements on a roller bearing mill in operation performed with a synchronizer”) since this tool gives the angular positions of the strains on the mandrel circumferentially. This technique is also optimal when permitting overloading of the mill, since the measured strains in the structure are a direct indicator of the loading on the machine.

The findings concerning this method give, in one way or another, answers to all four research questions.

**Models for calculation of charge parameters and loading on the mill**

The work involved in developing models for calculation of the charge parameters and the loading on the mill has resulted in a detailed study of the charge and the loading on the mill in operation. Formulas for calculations of the filling level, the charge torque, the charge mass, the position of the charge circumferentially, the loading, etc. are useful for many different types of analysis and investigations related to grinding mills. This part of the research work answers research questions 1 and 2.

**Load model and the MLC software**

The load model presented in this thesis is for quasi-static loading and the simulation of a full-size grinding mill. Due to the loading type, where the loading is applied as forces, accelerations and prescribed displacements, the simulations are computationally very effective, meaning low hardware requirements, in comparison, for example, with models based on distinct element methods, where the loading from the charge is simulated by motion and impacts of particles.
The presented load model is based on the assumption that the charge acts as a pure fluid. With this assumption, the normal and friction forces can be calculated based on the law of hydrostatic pressure. This means that the forces increase in linear proportion to the depth. This 100% fluid assumption entails some limitations. The real charge is not a pure fluid, but a mix of a fluid and solids, which creates a more vertical and downward-directed loading than that presented by the model (since solids mainly push in the direction of gravity); in the model the normal forces are directed normal to the surfaces due to hydrostatic pressure. For the same reason the forces on the heads are less in the real mill than in the model.

The assumption of the charge as a 100% fluid simplifies the calculation of the forces significantly. The load model can be adjusted to mimic the loading from the real charge more closely by taking the charge composition of a fluid and solids into account. The challenges, however, are to find the correct proportion for each phase, since the charge could be modelled as 25% solid and 75% fluid, 50% solid and 50% fluid, 75% solid and 25% fluid or some other mix that is unknown and varies constantly with production. The next question is how much this will influence the final results. If the charge composition exerts a great influence on the final results, it might be worthwhile to take it into account. This could be investigated in further work.

The MLC software combines the load model, the mathematical calculations for the charge configuration and various algorithms to calculate the forces and the loading acting on the mill in operation for the input of process and charge parameters. Based on the estimations for the load model, the MLC software gives very good and precise results.

The load model and MLC software give answers to research questions 1 and 2.

**Finite element modelling and numerical calculations**

Using the MLC software for the load calculation and the finite element model, the global displacement field of the entire mill can be calculated for any process situation. This answers research question 3. The most challenging part of the research work performed for this thesis was to develop the MLC-software and the finite element model with boundary conditions, spatial discretization, a material model, interactions and loading, without diverging too much from the real structural behaviour of the mill. To achieve even more precision in the calculated results, further refinement of the different parts of the models might be required.

Present day FEM calculations of large three-dimensional structures are often found to fall short of capturing real structural behaviour due to the complexity of the behaviour of real structures. The grinding mill is no exception, with even more difficulties being caused by the rotation of the mill and the dynamic and constant changing of the loading. Consequently, it is difficult to obtain accurate estimations of crack propagation speeds and critical crack sizes based on the calculated stresses.

Real life machines and structures seem to be less rigid than machines and structures as described by models. In the model, displacements can be prescribed to be zero by the use of constraints. In reality the displacements are never zero and small movements are always allowed everywhere.
For example, a bolt connection can be modelled as completely rigid by mathematical formulations, but in reality it is always more or less flexible. The larger a structure is, the greater is the extent to which such small movements are present, as a result of which the sum of these movements has a great influence on the behaviour of the complete structure. Since these small movements are not included in the model, the behaviour of the model and the real structure will be different in the end. Since more movements not being modelled are present in large systems, modelling such system is more complicated than modelling small systems. Moving systems with fluctuating loading are even more complicated to model than stationary ones with static loading.

**Verifications**

As mentioned earlier, the complexity of modelling the structural behaviour of a grinding mill in operation is high. Bearing this in mind, the agreement between the calculated and the measured strains and reaction forces at bearing supports has been good in the present research work. Complete agreement between measurements and calculated results is unrealistic when dealing with this type of problem since small estimations and simplifications of reality must be performed through the modelling.

The verification of models with real measurements is very important, especially when dealing with large three-dimensional structures, for the reasons explained above. Moreover, since the loading on the mill is constantly changing, the verification of results and measurements is necessary to obtain reliable results and to adjust the input parameters for the loading.

**Future research**

At this stage the global displacement field of the complete mill structure can be calculated for any input of process and charge parameters. Strain measurements and logged process data have been used to verify the calculated results with relatively good agreement.

A future step in this research is to use the global displacement field of the running mill to provide sub-model boundary conditions in detailed calculations aimed at fracture mechanics analysis and remaining fatigue life assessments. This can be accomplished by creating sub-models of the areas around the cracks and refining the finite element mesh significantly in these areas. The calculated displacements obtained from the full-size mill calculations can then be imported as prescribed displacements to the boundaries of the sub-models. With a special type of finite element code and crack elements, the stress intensity factor at the crack tips can be calculated for different crack lengths as the crack propagates. On the basis of Paris’ law and the material properties of the mill, the critical crack lengths, the crack propagation speeds and the time to failure can be calculated based on the mill speed, time in operation and loading.

The developed MLC software and the finite element model can in further work be used to investigate in detail how different charge and process parameters influence the mill strains in operation, with the purpose of finding the optimal process parameters, to avoid overloading and permit the safe running of the mill.
Furthermore, the MLC software and finite element model can be used in the product development of new mills, as they can be used in the design stage to evaluate the stresses in the mill and to optimize the mill design for different loadings. For this purpose the loading can be calculated for mean values of the process data over time and the peak loads for worst case scenarios.

References

Kyowa Electric Instruments Co., 2012, What’s a Strain Gauge? Introduction to Strain Gauges, Kyowa Electronic Instruments, Tokyo
Appended papers
Paper A

Structural analysis of a rotating grinding mill with the finite element method

Structural analysis of a rotating grinding mill with the finite element method

Filip Berglund

Division of Operation and Maintenance Engineering, Luleå University of Technology, SE 971 87 Luleå, Sweden, filip.berglund@ltu.se, +46-(0)920-49 38 20

Abstract

This paper presents a structural analysis performed on a rotating grinding mill using the finite element method, and the results from wireless strain measurements on a mill in operation. The displacement, strains and reaction forces have been determined for quasi-static loading. The circumferential and longitudinal strains have been measured on the outer mill mandrel surface.

In order to verify the computed results, the measured strain range for one complete rotation of the mill has been compared with the corresponding calculated strain range. The numerical results have also been verified with logged process data.

A mathematical model and a computer software have been developed for the calculation of the charge configuration, as well as the magnitude and distribution of the forces acting on the mill in operation.

Keywords: Rotating grinding mill; Finite element analysis; Wireless strain measurement; Computer software; Programming; Mill charge analysis; Mill load analysis.

1 Introduction

Rotating grinding mills usually work all around the clock, under heavy and fluctuating loads. The mill material undergoes alternating compression and tension load cycles as the mill rotates. Because of the severe working conditions, the most common failure type is fatigue.

Due to increasing production demands, the mills are today run with greater loads than when first installed. After many years in service, fatigue cracks and associated failures have recently begun to appear at an increasing rate in certain mills.

In order to obtain the global displacement field of the mill and to evaluate the stress in the critical areas for fatigue, a structural analysis based on the current loading is required.

Distinct element methods (DEM) have been used for a long time as a simulation tool to gain insight into particle flow processes. Cundall (1971) introduced DEM for analyses of rock mechanic problems (Jonsen et al., 2011). The method has been widely used to simulate the dynamic behaviour of the charge inside mills in operation (Raziszewski, 1999). One limitation of this method is that the computational costs increase significantly with the number of particles. Therefore, usually only two-dimensional (2D) models are presented or three-dimensional (3D) models with a limited extent in the axial mill direction. For the above-mentioned reason, DEM are seldom used to analyze the charge behaviour in a complete full-size mill. Because of the computational limits, the charge particles are often modelled as being too large and their shape is limited to spheres (Morrison and Cleary, 2004). Jonsen et al. (2011) describe how DEM can be used in combination with the finite element method (FEM)
to calculate strains in the mill liners and the underlying mill structure in a 0.1 m long cross-section part of a pilot mill in operation.

The charge behaviour in the mill under investigation is relatively mild; i.e. hardly any spin or turbulence of the charge occurs and the dynamic impact from falling particles is insignificant. Consequently, it is suitable to use a quasi-static simulation approach instead of a dynamic one like DEM. The quasi-static approach requires much less computational resources than the DEM solution and the complete mill structure can be analyzed with a minimum of computational costs. For this reason, the quasi-static approach was used for the structural analysis of the mill. A structural analysis of this kind has recently been reported for a rotary kiln (Del Coz Diaz et al., 2002), but similar works on mills are scarce.

In order to obtain the quasi-static loading from the charge in the running mill, it is necessary to know the shape, volume, position, mass and torque of the charge. Bond (1960) describes the foundations of the torque-arm model for mill power. This model describes analytically the shape and position of a non-centrifuged mill charge in a running mill. The torque-arm model uses a chord connecting the toe and shoulder of the charge to delineate the charge shape. The model does not work as well at high mill speeds as it does at low speeds (Dong and Moys, 2003).

If a large amount of slip occurs between the outer layer of the charge and the liners (or if the mill speed is low, for example less than 60 % of the critical speed), the charge attains the shape shown in Figure 3, and the torque-arm model for mill power is fairly accurate (Moys, 1993; Moys et al., 1996). This model works well for describing the charge in the mill under investigation due to its specific low lifter height, typical charge composition, mill speed and mild charge behaviour.

Tano (2005) describes the use of a strain gauge mounted in a lifter to measure different charge features, such as the filling level and the behaviour and position of the charge. However, there is a lack of literature dealing with the utilisation of strain gauges to measure the strains on the outer mill mandrel on a running mill. Measurements of this type have been performed by the author on the mill under investigation in order to verify the computed strains.

The structural analysis in this paper concerns the second of two pebble mills in a production line situated at a mining company in northern Sweden. The analysis has been performed with the finite element method (Zienkiewicz and Taylor, 2000) and the numerical results have been compared with in situ strain measurements on the mill in operation and with logged process data.

The grinding mill charge is a complex multi-phase system with a range of particle and material processes that varies with time and depends on the exact operating conditions of the mill. The operating conditions generally vary stochastically over time (Nierop et al., 2001). Consequently, the uncertainty of the real charge distribution and the fact that its behaviour is difficult to monitor make it necessary to investigate different charge distributions and load cases in order to find the most realistic and suitable one for the present situation.

As a first step in the analysis, the charge behaviour and loading conditions were studied in detail in order to determine the shape, volume, position, mass and torque of the charge and the size and distribution of the forces acting on the mill during operation. For this purpose a mathematical model and a computer software, named the Mill Load Calculator (MLC), were developed by the author.

A mill model including the linings, discharger and surrounding components was created and discretized into a finite element mesh. Due to the presence of geometric transitions in the model, 3D solid elements were used. A mesh convergence study was performed where the mesh was refined in several steps.

After finding the most suitable mesh and structure constraints, quasi-static and linear elastic small deformation calculations of the displacement field were performed. From the
displacement field, the structural strains and reaction forces were obtained with standard routines.

For each load case, the strains in the circumferential and the longitudinal direction at the position of the applied strain gauges were plotted for one whole 360° rotation of the mill. The calculated maximal strain range for each 360° curve was compared to the corresponding measured one. For further verification, the calculated reaction forces at the bearing supports were compared to the corresponding logged ones obtained from bearing pressures.

2 Description of the actual physical mill

The mill under investigation is a secondary pebble mill of the grate discharging type, supported by hydrostatic pressure bearings. The diameter of the mill is 6.5 m and the length 8.5 m. The rotation speed of the mill is fairly constant at 12.75 rpm (revolutions per minute), which is approximately 75% of the critical speed. The critical speed is defined as the rotational speed at which the centrifugal forces equal the gravitational forces at the mill mandrel's inside surface. This is the rotational speed at which particles will not fall away from the mill's mandrel (Rose and Sullivan, 1957; Watanabe, 1999).

The iron ore material, together with a large amount of water, enters the mill at the inlet end. The ore is crushed and ground to a fine powder inside the mill, in a process where larger ore lumps are grinding smaller ones. At the outlet end, ore with a small enough lump size passes through a grate and is lifted out of the mill via a discharge wheel. The ore is then poured into a trommel sieve, beyond which the finest powder grades are further enriched and processed. The function of a discharge wheel is described by Latchireddi and Morrell (2003).

The mill is, in plain terms, a rotating cylindrical barrel with three main parts: the mandrel and the inlet and outlet heads. The heads are circular plates attached to each end of the mandrel, see Figure 13.

The inside surface of the mill is covered with magnetic linings, mainly for protection of the mill surface. The mill is equipped with two steering wheels, one at each end, supported by three hydrostatic pressure bearings preventing radial displacement of the mill. Axial movement of the mill is prevented by two vertical pressure bearings at the outlet end, one for each direction. The bearings are separated from the mill by a thin oil film, which provides almost frictionless contact. The pressures on the main load supports are continuously logged.

The mill is driven by a single electrical motor, via a pinion, acting on a tooth drive wheel attached to the mill mandrel. The maximum motor power is 4,500 kW. The motor power used varies and is frequently logged. The mill is made of ordinary construction steel whose properties according to the manufacturer are shown in Table 1.

Table 1. Material properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Shell, inlet and outlet head</th>
<th>Drive wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of elasticity [MPa]</td>
<td>210 000</td>
<td>167 000</td>
</tr>
<tr>
<td>Poisson’s ratio [-]</td>
<td>0.29</td>
<td>0.29</td>
</tr>
<tr>
<td>Yield strength [MPa]</td>
<td>220</td>
<td>220</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>7800</td>
<td>7200</td>
</tr>
</tbody>
</table>

Figure 1 shows the outlet end of the mill, and the trommel sieve is located inside the orange steel box visible in the picture.

For more information about the function of the specific mill type, see Wills et al. (2006).
3 Strain measurements

In order to understand the loading on the running mill, to verify the calculated strains and to obtain knowledge about the strains in the mill structure, strain measurements were performed on the mill during operation. The measurements were carried out by the author.

Because of the rotating machinery, a wireless measurement system was used. The system has a logger with a receiver connected to two strain gauges, which are fixed at the same position on the mill. The logger, powered by a rechargeable battery, is controlled wirelessly by a transmitter connected to a computer. With this setup, the measurements can be triggered and stopped remotely at any time.

The high costs of production stoppages make it difficult and very expensive to stop the mill in an unplanned manner. For this reason, the equipment was attached to the mill during a maintenance stoppage. The measurements were then performed after the mill had been restarted and the production had been stabilized. The remote-controlled system was indispensable for the possibility of triggering measurements on the running mill.

An L-shaped strain sensor consisting of two strain gauges perpendicular to each other was attached to the outside of the mandrel, at a position close to the mid-section of the mill 1.65 m from the flange where the drive wheel is attached. One strain gauge was oriented circumferentially and the other longitudinally. The exact position of the strain gauges was registered in order to allow comparison with FEM-computed results for the same position.

The strains were measured with a frequency of 1,024 Hz during several mill rotations. The results of one measurement are shown in Figure 2, where the strains are plotted against time. As can be seen in Figure 2a, the strain pattern is similar and repetitive, with almost the same strain range in each cycle. Each repetitive pattern corresponds to one 360° rotation of the mill. In Figure 2b the measured strains for a single mill rotation are shown.

The strain gauges were fixed and calibrated on the mill on an occasion when the mill was idle and still loaded, and therefore strains already existed in the structure. Consequently, the obtained results show only the variation of the strains during each rotation and not the absolute values of the strains in the structure.

Seen over a longer period of time, the mill is subjected to fatigue loading of variable amplitude because of the characteristic changes in the process and charge parameters, e.g. the engine power, charge mass, filling level, etc. However, seen over a short period, the measurements indicate that the mill is subjected to fatigue loading of almost constant amplitude. Because of the short-term constant loading, a quasi-static analysis was performed for the calculation of the strains in the mill at a certain instant.

The strain cycles in Figure 2 drift slightly upwards with regard to the strain level over time, which is probably due to decreasing battery voltage in the logger during the measurement.
Figure 2. Measured strains in the circumferential and the longitudinal direction: (a) curves for several mill rotations, (b) curves for one single rotation

4 Charge and loading analysis

For the structural analysis, the loads acting on the running mill must be known. In order to determine the loads due to the charge, the distribution and volume of the charge inside the running mill must be calculated, as well as the total weight of the charge mass and the torque of the charge.

In the following section, two models are presented. The first describes the charge position and shape in a mill in operation and the loads acting on the running mill, while the second calculates the magnitude and distribution of the loads acting on the mill in operation.

4.1 Charge analysis

The charge consists of ore lumps of varying size, typically 35 mm in diameter and less. The charge also contains a large amount of water. In operation the upper surface of the axial cross-section of the charge is typically concave; see Cleary (1998) and (2001). However, at first the upper surface is assumed to be straight and, based on this assumption, the charge takes the shape of a circle segment, see Figure 3. It is further assumed that the charge is uniformly distributed geometrically and with regard to its density along the mill.

The position of the charge circumferentially is obtained from the power equilibrium of the mill and motor. During operation the motor power rotates the charge to one side, see Figure 3. The displacement of the charge, from the zero torque position, is a function of the motor power, the mill rotation speed, the total charge weight and the filling level inside the mill.
The power supplied to the electric motor is transferred through a simple gear box, via a pinion, to the mill itself. According to the law of energy conservation, the power equilibrium equation of the mill and the motor can be written as:

\[ P_{\text{mill}} = P_{\text{motor}} \cdot \eta \]  

(1)

where \( P_{\text{mill}} \) is the power of the mill and \( P_{\text{motor}} \) is the power of the motor. \( \eta \) is a factor taking into account the loss of power in each gear transmission through the drive-train and is here set to 0.98. \( n \) is the total number of gear transmissions, here given as 3.

The mill itself is a large rotational system and the power of the mill can be written as:

\[ P_{\text{mill}} = M_{\text{mill}} \cdot \omega \]  

(2)

where \( M_{\text{mill}} \) is the mill torque and \( \omega \) is the angular velocity of the mill. From equation (1) and (2), \( M_{\text{mill}} \) can be obtained as:

\[ M_{\text{mill}} = \frac{P_{\text{motor}} \cdot \eta}{\omega} \]  

(3)

During the operation of the mill, the charge is positioned to one side. The torque of the charge, \( M_{\text{charge}} \), can then be calculated as:

\[ M_{\text{charge}} = d(H, \alpha) \cdot m \cdot g \]  

(4)

where \( d \) is the horizontal distance between the mill centre and the centre of gravity (CG) of the charge, see Figure 3. \( m \) is the total equivalent mass of the charge, i.e. the part of the total charge carried directly by the mill, but not free-flying particles or lumps. \( g \) is the acceleration due to gravity. \( d \) is obtained by the sine law as:
\[ d = S \cdot \sin(\alpha) \]  

(5)

where \( S \) is the radial distance between the mill centre and the CG of the charge. \( \alpha \) is the angular displacement of the charge’s CG counterclockwise with respect to a vertical line through the centre of the mill. \( S \) is calculated according to the formula:

\[ S = \frac{4 \cdot r \cdot \sin^3(\beta/2)}{3 \cdot (\beta - \sin(\beta))} \]  

(6)

and \( \beta \), which is the segment angle in radians, is calculated according to the cosine law as:

\[ \beta = 2 \cdot \arccos \left( \frac{r - H}{r} \right) \]  

(7)

Here \( H \) is the maximum radial height of the charge circle segment and \( r \) the inner radius of the mill. By inserting equation (7) and (6) into (5), the distance \( d \) can be expressed as:

\[ d(H, \alpha) = \frac{4 \cdot r \left(1 - \left(\frac{r - H}{r^2}\right)^2\right)^{3/2}}{6 \cdot \arccos \left(\frac{r - H}{r}\right) - 3 \cdot \sin \left(2 \cdot \arccos \left(\frac{r - H}{r}\right)\right)} \cdot \sin(\alpha) \]  

(8)

By inserting (8) into (4), \( M_{\text{charge}} \) can be calculated as:

\[ M_{\text{charge}} = \frac{4 \cdot r \left(1 - \left(\frac{r - H}{r^2}\right)^2\right)^{3/2}}{6 \cdot \arccos \left(\frac{r - H}{r}\right) - 3 \cdot \sin \left(2 \cdot \arccos \left(\frac{r - H}{r}\right)\right)} \cdot \sin(\alpha) \cdot m \cdot g \]  

(9)

The friction between the mill and the hydrostatic bearing support is insignificant and is therefore neglected. Making this assumption, the torque of the charge equals the mill torque:

\[ M_{\text{charge}} = M_{\text{mill}} \]  

(10)

Now, inserting equation (3) and (9) into (10) and restructuring give \( \alpha \) as:

\[ \alpha = \arcsin \left( P_{\text{motor}} \cdot \eta \cdot \frac{6 \cdot \arccos \left(\frac{r - H}{r}\right) - 3 \cdot \sin \left(2 \cdot \arccos \left(\frac{r - H}{r}\right)\right)}{4 \cdot \omega \cdot m \cdot g \cdot r \left(1 - \left(\frac{r - H}{r^2}\right)^2\right)^{3/2}} \right) \]  

(11)

During production, \( \omega, g, \eta, n, \) and \( r \) are constants, while \( H, m \) and \( P_{\text{motor}} \) vary with time, and therefore \( \alpha \) varies with time. \( m \) and \( P_{\text{motor}} \) are continuously logged, so only \( H \) is unknown during production. Equation (11) gives the relationship between \( \alpha \) and the parameters.
mentioned during operation and can be used to calculate the charge distribution inside the mill at any given time or in any given process situation.

For grinding mills, it is more common to describe $H$ in terms of a relative filling level, $V$, defined as the ratio between the charge volume and the total available volume inside the mill. Because of the assumption of equal axial charge distribution, $V$ can be written as:

$$ V = \frac{A_{\text{charge}}}{A_{\text{mill}}} \tag{12} $$

where $A_{\text{charge}}$ is the cross-section area of the charge and $A_{\text{mill}}$ the cross-section area of the mill. The expressions for these are obtained as:

$$ A_{\text{charge}} = \frac{r^2}{2} \cdot \left( \beta - \sin(\beta) \right) \tag{13} $$

$$ A_{\text{mill}} = \pi \cdot r^2 \tag{14} $$

By inserting equation (7), (13) and (14) into (12), the filling level can be expressed as:

$$ V(H) = \frac{\arccos \left( \frac{r - H}{r} \right) - \frac{1}{2} \cdot \sin \left( 2 \cdot \arccos \left( \frac{r - H}{r} \right) \right)}{\pi} \tag{15} $$

4.2 Filling level and charge density during the strain measurement

The charge of the mill moves continuously when the mill is in operation. The charge mass, filling level and supplied power all vary during production. To be able to compare the computed and the measured strains, the charge distribution and density need to be determined for the specific strain measurement period.

The parameters $V$ and $\alpha$ describe the relative filling level and the angular displacement of the charge inside the mill and need to be calculated for the measurement period. In equation (11) and (15), the parameters $\alpha$, $g$, $\eta$, $n$, and $r$ are known and, by inserting the logged values of $m$ and $P_{\text{motor}}$, only one of the parameters $\alpha$ and $V$ needs to be estimated in order to calculate the other.

During the measurements, $m$ was calculated to be 372.1 tons from the logged bearing pressures by subtracting the mass of the empty mill and installed components from the total mass logged on the bearings during the measurement period. The detailed procedure to calculate $m$ from the bearing pressures is described in Appendix D. For the same period, $P_{\text{motor}}$ was ascertained to be 4,095 kW based on logged data. Now the charge torque, $M_{\text{charge}}$, during the measurement period can be calculated from equation (3) and (10).

In order to understand the relation between $M_{\text{charge}}$, $V$ and $\alpha$ for the logged value of $m$, $M_{\text{charge}}$ was calculated, using equation (9) and (15), for $V$ in the range 0-100% and $\alpha$ in the range 0-90º, see Figure 4. For any given $m$, the maximal $M_{\text{charge}}$ is obtained at $\alpha = 90^\circ$ and a minimum of $V$. Increasing $V$ leads to decreasing $M_{\text{charge}}$ and increasing $\alpha$, up to 90º, leads to increasing $M_{\text{charge}}$. In simulations of operation, however, $M_{\text{charge}}$ is subject to the equilibrium condition:

$$ M_{\text{charge}} = M_{\text{logged}} \tag{16} $$
$V$ is obtained from the condition that only combinations of $H$ and $\alpha$ in equation (9) satisfying equation (16) are admissible. In Figure 4, the corresponding values of $V$ and $\alpha$ are situated on the surface line $M_{\text{charge}} = M_{\text{logged}}$ (the value 100% in the plot). Only the combinations of $V$ and $\alpha$ corresponding to this surface line are treated further, see Figure 4b.

![Figure 4. Relation between $M_{\text{charge}}$, $V$ and $\alpha$ for the constant $m$ of 372.1 tons: (a) 3D plot, (b) cross-section at $M_{\text{charge}} = M_{\text{logged}}$, (c) cross-section at $\alpha = 30.78^\circ$, (d) cross-section at $V = 40\%$](image)

$V$ and $\alpha$ are both unknown parameters and either can be calculated from an estimate of the other. $V$ was chosen for estimation, because more information is available for this parameter. According to the mill and process specifications provided by the mining company, $V$ should be in the range of 30-45%. If $V$ is greater than 45%, this leads to back-flow in the mill. The range of $V$ is therefore known roughly, but a more accurate estimation is desirable in the calculations. This is performed by comparing the theoretical value of the charge density, $\rho_{\text{theory}}$, with the charge density, $\rho$, calculated from $V$ and the calculated value of $m$. Because of the assumption of a uniform axial charge distribution, $\rho$ is calculated as:

$$\rho = \frac{m}{V \cdot L \cdot \pi \cdot r^2}$$  \hspace{1cm} (17)

It is noted that $\rho$ is a function of $m$ and $V$.

The charge consists of crushed magnetite mixed with a large amount of water. Based on this knowledge, $\rho_{\text{theory}}$ was calculated with the following equation:

$$\rho_{\text{theory}} \approx \rho_{\text{magnetic solid}} \cdot \gamma_{\text{crack}} \cdot \gamma_{\text{water}} = 3600 \text{ kg/m}^3$$  \hspace{1cm} (18)

Here $\rho_{\text{magnetic solid}}$ is the density of solid magnetite, which is in the range 4,900-5,200 kg/m³. Due to the fact that a certain amount of the charge is gangue minerals, a lower value of 5,000
kg/m³ was used for the calculation. \( \gamma_{\text{crush}} \) is a reducing factor for the crushing with a value of 0.6 and \( \gamma_{\text{water}} \) is a magnifying factor for the effect of water with a value of 1.2. The factors for solid, crushed and water-filled densities for a number of rock types were obtained from several web sources, see the last ones in the list of references. The calculated charge density was verified with the mining company and is considered to be within the range of normal production.

In order to find the charge density closest to \( \rho_{\text{theory}} \), \( \rho \) was calculated by equation (17) for filling levels between 30% and 45% and the calculated value of \( m \), see Table 2. The corresponding \( \alpha \) for each \( V \) was calculated by equation (11) and (15). In order to visualize the obtained charge distributions and to check whether they were realistic, the charge profiles were plotted for \( V = 30, 35, 40 \) and 45%, see Figure 5. The inner circle in Figure 5 represents the inlet/outlet hole of the mill.

<table>
<thead>
<tr>
<th>( V ) [%]</th>
<th>( \alpha ) [º]</th>
<th>( \rho ) [kg/m³]</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>24.14</td>
<td>4760</td>
</tr>
<tr>
<td>32.5</td>
<td>25.18</td>
<td>4394</td>
</tr>
<tr>
<td>35</td>
<td>26.29</td>
<td>4080</td>
</tr>
<tr>
<td>37.5</td>
<td>27.49</td>
<td>3808</td>
</tr>
<tr>
<td>40</td>
<td>28.80</td>
<td>3570</td>
</tr>
<tr>
<td>42.5</td>
<td>30.22</td>
<td>3360</td>
</tr>
<tr>
<td>45</td>
<td>31.78</td>
<td>3174</td>
</tr>
</tbody>
</table>

Figure 5. Charge profiles with the CG marked for different \( V \): (a) \( V = 30\% \), (b) \( V = 35\% \), (c) \( V = 40\% \), (d) \( V = 45\% \)

Here, the filling level \( V = 40\% \) yields a charge density very close to \( \rho_{\text{theory}} \). On the basis of this and a comparison of the charge profiles, \( V = 40\% \) and \( \rho = 3,570 \) kg/m³ were selected for the load analysis.

### 4.3 Loading analysis

In order to calculate the loading on the mill, the charge is assumed to act like a fluid exerting pressure and friction on the mill. Based on this assumption, the loads on the mill are the following:

- Normal forces (\( F_{\text{normal}} \)) from the charge, each acting on an element of the area of the mandrel and of the heads.
- Friction forces (\( F_{\text{friction}} \)) from the charge, each acting on an element of the area of the mandrel and of the heads.
- Tangential forces (\( F_{\text{wheel}} \)) and normal forces (\( F_{\text{wheel normal}} \)), from the pinion, acting on the drive wheel. Because of the helical gearing, there are also force components in the axial mill direction (\( F_{\text{wheel axial}} \)).
Gravity forces corresponding to the weight of the mill itself and the attached components (including the trommel sieve, attached linings, discharger, etc.).

Radial and axial reaction forces at the bearing supports ($F_{bearing}$ and $F_{bearing\ axial}$), the latter at the outlet end only.

Figure 6 shows simplified free body diagrams of the forces acting on the mill during operation. The green arrows represent the forces acting on the mill heads and the red arrows the forces acting on the mandrel. Friction forces between the mill and the hydrostatic bearings are neglected.

During operation, in addition to equation (10), the following torque equilibrium condition exists for the mill:

$$M_{charge} = M_{friction} = M_{wheel} \quad (19)$$

where $M_{friction}$ is the total torque from the friction forces acting on the mill from the charge, $M_{wheel}$ is the total torque from the tangential forces acting on the drive wheel, and $M_{charge}$ is given by equation (9) and (10).

Based on the knowledge of the charge distribution, gear data, $m$, $M_{logos}$, and equation (19), the magnitude and distribution of the forces acting on the mill, as well as the coefficient of friction between the charge and the mill ($\mu$), can be calculated.

The procedure for calculating the forces from the charge can be found under the section “Charge load calculation algorithm” below, and the procedure for calculating the forces acting on the drive wheel can be found in Appendix E.
4.4 Variations in the charge distribution

So far the top surface of the charge has been assumed to be plane and the axial density distribution to be uniform. In reality, the top surface is more or less concave and the density distribution slightly non-uniform axially. The charge and the load calculations are now adjusted accordingly.

The radius of the top surface curve, assumed to be circular, is denoted by \( R \), see Figure 7a. The equations and derivations for the calculations of the charge torque and filling level for the charge with a concave upper surface are given in Appendix A.

The charge inside the mill is inhomogeneous and mainly consists of two different and more or less mixed phases, namely pebbles and pulp. The pebble phase, about 60% of the charge volume, consists of larger iron pieces, of different size, typically in the range of 6-30 mm in diameter. The pulp phase, the remainder, consists of a mix of water and very fine iron pieces. The pebbles act as a grinding medium and the pulp is the outcome from the mill. The exact amount of each phase is dependent on the instantaneous process situation.

With regard to volume, the charge distribution is fairly uniform in the axial mill direction, but the mix of the different phases is not. The concentration of pulp is greatest near the inlet end and decreases linearly towards the outlet end, where the pulp (not the concentration) is lifted out of the mill. The non-uniform charge density distribution along the mill is here assumed to be linear and is obtained from different bearing pressures at the in- and outlet ends, see Figure 7b.

4.5 Mill load calculating software

In order to calculate the charge distribution and the loads acting on the mill for different charge and process parameters, a computer software called the Mill Load Calculator (MLC) was developed by the author.

The software calculates the charge distribution, \( \alpha, m, \rho, M_{\text{charge}} \), the force distributions, \( F_{\text{normal}}, F_{\text{friction}} \), and the forces \( F_{\text{wheel}}, F_{\text{wheel normal}} \) and \( F_{\text{wheel axial}} \) from the input of the bearing pressures, \( \alpha, V, P_{\text{motor}} \), and \( R \). The program also delivers the present trommel mass, \( m_{\text{trommel}} \), the specific pebble and pulp density, \( \rho_{\text{pebbles}} \) and \( \rho_{\text{pulp}} \), and the coefficient of friction between the charge and the mill, \( \mu \).

The bearing pressures for the main bearings are denoted, according to the standard of the mining company, by \( P_{112}, P_{113}, P_{114}, P_{115}, P_{116} \) and \( P_{117} \), see Figure 9.

In addition to the above-mentioned outputs, the software also visualizes the charge distribution, along with the position of the charge’s CG, in 2D and 3D, together with the distributions of the forces acting on the mill, in the graphical user interface, see Figure 9.

The axial charge mass distribution is visualized with two bars in Figure 9, and each bar indicates how much of the total charge mass is portioned at each mill end. The charge surface of the 3D plot has a faded colour, and there is a darker colour where the charge density is higher. This indicates further the axial charge density distribution inside the mill.
The calculated loads and their distributions are exported to the finite element software. The computational mill model is updated and a solution is calculated for the current load case. The data transfer is platform-independent. Figure 8 shows a block diagram of the calculation workflow including the MLC software. Based on the MLC software data, the finite element model yields the stresses, strains, reaction forces, displacements, etc. in the mill for any given input of process and charge parameters.

Figure 8. Block diagram of the calculation workflow

Figure 9. Graphical user interface for the MLC software
4.6 Charge load calculation algorithm

Since no commercial FEM software found included any specific routine for calculating and distributing the forces from the charge in a mill in operation, an algorithm for this purpose was developed by the author and embedded into the MLC software. The basis for the algorithm is explained below.

![Figure 10. Areas for force action from the charge on the mill mandrel and heads](image)

The form and orientation of the charge inside the running mill are given by the parameters $V$, $\alpha$, and $R$. The area where the charge forces act is denoted by $S_{\text{mandrel}}$ for the mill mandrel and $S_{\text{head}}$ for the mill heads. $S_{\text{head}}$ is the total mill head area in contact with the charge (coloured light blue in Figure 10). $S_{\text{mandrel}}$ is the total mill mandrel area in contact with the charge between $\Phi = 180^\circ$ and $\Phi = 360^\circ$, see Figure 10.

In order to obtain the forces from the charge, the surfaces $S_{\text{mandrel}}$ and $S_{\text{head}}$ are divided into a large number, $m$ and $n$, respectively, of discrete area elements, each with an area denoted by $A_{\text{mandrel}}$ and $A_{\text{heads}}$, respectively. The position of each area element is calculated. The pressure from the charge acting on an area element is given by the hydrostatic law:

$$p_{\text{element}} = \rho \cdot g \cdot h$$  \hspace{1cm} (20)

where $h$ is the vertical charge height above the element. The normal force acting on the element can then be written as:

$$F_{\text{normal element}} = p_{\text{element}} \cdot A = \rho \cdot g \cdot h \cdot A$$  \hspace{1cm} (21)

where $A$ is the area of the element. Since $h$ is calculated vertically from a surface element and upwards, no forces are acting on the mandrel between $\Phi = 0^\circ$ and $\Phi = 180^\circ$, see Figure 10.

The friction force acting on an area element is obtained by multiplying the normal force acting on the element by the coefficient of friction $\mu$ between the charge and the mill:

$$F_{\text{friction element}} = \mu \cdot F_{\text{normal element}}$$  \hspace{1cm} (22)
As can be seen in equation (21), the magnitude of the normal and the friction force at a given position is directly proportional to \( h \) at the position in question. The torque from friction on an area element is given as:

\[
M_{\text{element}} = F_{\text{friction element}} \cdot r_{\text{element}}
\]

(23)

where \( r_{\text{element}} \) is the radius to the element. The total torque from the friction forces acting on the mill is the sum of the products of the friction force acting on an area element and the corresponding radial distance from the centre of the mill:

\[
M_{\text{friction}} = \sum M_{\text{element}} = \mu \cdot \rho \cdot g \cdot \left( A_{\text{mandrel}} \cdot r \cdot \sum_{i=1}^{n} h_i + 2 \cdot A_{\text{head}} \cdot \sum_{j=1}^{m} h_j \cdot r_j \right)
\]

(24)

In this equation, \( h_i \) is the vertical charge height above the area element at position \( i \) on \( S_{\text{mandrel}} \) and \( h_j \) is the vertical charge height above the area element at position \( j \) on \( S_{\text{head}} \). \( r_j \) is the radius to the area element at position \( j \) on \( S_{\text{head}} \), see Figure 10. Rewriting equation (24) gives the coefficient of friction between the charge and the mill as:

\[
\mu = \frac{M_{\text{friction}}}{\rho \cdot g \cdot \left( A_{\text{mandrel}} \cdot r \cdot \sum_{i=1}^{n} h_i + 2 \cdot A_{\text{head}} \cdot \sum_{j=1}^{m} h_j \cdot r_j \right)}
\]

(25)

where \( M_{\text{friction}} \) is known from equation (19). Inserting equation (25) into (22) gives the friction force at any location.

From the algorithm and the equations described above the normal and friction force distributions from the charge are calculated with hundreds of thousands of forces. The total number of forces can be set to any value and is only limited by the computer capacity.

The distributions of the area element’s normal and friction forces are described by the coordinates for the discrete positions of the forces over \( S_{\text{mandrel}} \) and \( S_{\text{head}} \). The force distribution between the discrete points is interpolated in the FEM-software.

The information about the magnitude and distribution of the forces is stored in platform-independent files which are used for the communication between the MLC software and any finite element software.

In Appendix B and C are shown comparisons between the MLC algorithm and the corresponding hand-calculations of the magnitude of a normal force at a known position on the mill head.

### 4.7 Calculated load cases

The MLC software was used to calculate the loads acting on the mill during the strain measurement period. For this calculation, all the input parameters were known except \( R \). The bearing pressures and \( P_{\text{motor}} \) are logged, \( \alpha \) is constant, \( V \) is obtained as explained in Section 4.2, and the axial charge density distribution is obtained from the difference between the bearing pressures of the inlet and the outlet end.

As the parameter \( R \) is unknown, different values of this parameter were assumed in order to investigate its influence on the final results. A total of 6 different charge distributions and
load cases were calculated by the MLC software. The input parameters to the MLC software for each load case are shown in Table 3 and the obtained results are shown in Table 4.

Figure 11 and Figure 12 show the charge distributions visualized graphically for the load cases. The inner circles in Figure 11 and Figure 12 represent the inlet/outlet holes of the mill.

### Table 3. Input parameters to the MLC software for calculation of the load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Input parameters</th>
<th>Variation</th>
</tr>
</thead>
<tbody>
<tr>
<td>L1</td>
<td>Inf (no concavity)</td>
<td>6928</td>
</tr>
<tr>
<td>L2</td>
<td>P112=54.64 bar, P113 =54.21 bar</td>
<td>4405</td>
</tr>
<tr>
<td>L3</td>
<td>P114=46.35 bar, P115=44.57 bar</td>
<td>3499</td>
</tr>
<tr>
<td>L4</td>
<td>P116=47.03 bar, P117=51.85 bar</td>
<td>2999</td>
</tr>
<tr>
<td>L5</td>
<td>V=40 %, P_motor=4095kW, ( \omega =12.75 ) RPM</td>
<td>2704</td>
</tr>
</tbody>
</table>

### Table 4. Output parameters from the MLC software for calculation of the load cases

<table>
<thead>
<tr>
<th>Load case</th>
<th>Output parameters</th>
<th>Variations</th>
</tr>
</thead>
<tbody>
<tr>
<td>L1</td>
<td>m=372.1 tons, ( m_{trommel}=11.6 ) tons</td>
<td>0.161 28.80</td>
</tr>
<tr>
<td>L2</td>
<td>( \rho_p=3570 ) kg/m(^3), ( \rho_{pebbles}=4200 ) kg/m(^3)</td>
<td>0.148 29.05</td>
</tr>
<tr>
<td>L3</td>
<td>( \rho_{steel}=2626 ) kg/m(^3), M_char=2887 kNm</td>
<td>0.138 29.57</td>
</tr>
<tr>
<td>L4</td>
<td>F_wheel=649.8 kN</td>
<td>0.129 30.32</td>
</tr>
<tr>
<td>L5</td>
<td>F_wheel_normal=242.5 kN</td>
<td>0.128 31.55</td>
</tr>
<tr>
<td>L6</td>
<td>F_wheel_axial=85.2 kN</td>
<td>0.127 33.66</td>
</tr>
</tbody>
</table>

Figure 11. Charge profiles with the CGs marked for load cases (a) L1, (b) L2, (c) L3, (d) L4, (e) L5, and (f) L6

Figure 12. 3D visualization of charge, (a) load case L3, (b) load case L5
5 Finite element model

This section describes the finite element model of the mill. Because of the non-uniform loading conditions, no symmetries prevailed and a full model analysis was performed. Because of the assumed small deformations in the mill, geometrical non-linear analysis was not considered necessary. Therefore, a purely linear elastic material model was used. The modelling was performed with Siemens’ PLM Software, UGS NX7.5 (NX 7.5 Help and Documentation, 2010).

5.1 Geometry

The first step involved the creation of a computer model of the mill from drawings. The computer model was simplified in order to facilitate discretization into finite elements and to reduce the computational time, Figure 13.

The simplified model included only those details which were deemed relevant to the structural stiffness; e.g. bolt holes, small corner radii and similar details were neglected. A major simplification was obtained by neglecting the holes in the drive wheel web. The mass of the wheel was maintained, as its weight is carried by the mill. The drive wheel could be greatly simplified because this part is not intended for analysis but mainly used to maintain accurate weight and for the transfer of force from the pinion to the mill mandrel.

At this stage the material properties presented in Table 1 were applied to the model.

5.2 Spatial discretization

The simplified mill geometry was discretized into a finite element mesh with 3D (three-dimensional) solid brick elements, see Figure 14. The 3D mesh was created by sweeping a 2D seed mesh around the mill’s axis of revolution. The ideal element length-width-height ratio is 1:1:1 and the most extreme such ratio in the model is 11:10:1. Elements with an extreme ratio are, however, situated far from the point of interest. The trommel mass was modelled as a 0D point mass element. The point mass was connected to the mill outlet end using 1D spider elements. This creates a ridged connection between the point mass and the mill, transferring gravity forces and bending torque to the mill. Figure 15 and Figure 16 show cross-sections of the mesh.
5.3 Boundary conditions

The mill is supported in radial directions at each end and axially at the outlet end by the hydrostatic pressure bearings. In practice the supports allow small displacements. The boundary conditions representing the bearings were modelled as constraints applied to the nodes at the contact areas of the bearings, see Figure 17. For the six bearings carrying the main load and preventing the movement of the mill in the radial directions, the constraints
were locked in the radial direction and free in all the other directions. For the axial bearings, the constraints were locked in the axial direction only.

In order to eliminate computational singularity in the model stiffness matrix, the model was locked for rotation by a constraint with a prescribed displacement set to zero in the angular direction of the mill axis only. The constraint was applied in one node at the bottom of the steering wheel at the outlet end, see Figure 17.

![Figure 17. Applied boundary conditions](image)

### 5.4 Convergence with mesh refinement

It is important to obtain accurate results, particularly in the area of the strain gauges. For this reason, the number of elements was increased locally in the radial, circumferential and axial directions around the entire circumference of the mill at the axial position of the strain gauges. See the local mesh refinement zone in Figure 18.

To reduce the total number of degrees of freedom to a minimum and to save computation time, it is important to use as few and as large elements as possible. At the same time, the use of too large elements yields less accurate results. In order to find the largest permissible element size which can be used without compromising the accuracy, a convergence study was performed in which the mesh was refined locally and globally in several steps. For each mesh the node strains in the longitudinal and circumferential directions, around the circumference on the outer surface of the mandrel, were plotted in the local refinement zone, see Figure 20a. The starting point for the plots is according to Figure 3.

One part of this study involved testing two different 3D element types in order to find the most appropriate one for the application: 8 node linear approximation and 20 node quadratic approximation.

In Table 5, details of the investigated meshes are shown. For all calculations during the convergence study, load case L1 was used.

Figure 20b shows the maximum strain in the circumferential and longitudinal directions plotted against the total number of nodes for each mesh. From Figure 20 it is found that the strain curves are similar for most meshes, and that only meshes E1, E7 and E9 deviate from the general pattern. It must be noted that these meshes have comparatively few elements circumferentially.

The structure of the mesh with 20 node quadratic elements, E9, is very similar to that of mesh E1 with its 8 node linear elements, despite the difference in element type between the
two meshes. The strain curves for these meshes are about the same and the conclusion is that both linear and quadratic elements can be used.

Based on the results of this study, mesh E4 was used in further simulations. With this mesh the convergence point is reached. This mesh is located at the point where the curves in Figure 20b start to level off.

Table 5. Investigated finite element meshes

<table>
<thead>
<tr>
<th>Mesh</th>
<th>Global refinement</th>
<th>Element type</th>
<th>Total number of nodes</th>
<th>Longitudinal direction</th>
<th>Radial direction (thickness)</th>
<th>Circumferential direction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>8 node 20 node</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>3D linear 3D quadratic</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>E1</td>
<td>No</td>
<td>X</td>
<td>148 669</td>
<td>6</td>
<td>3</td>
<td>156</td>
</tr>
<tr>
<td>E2</td>
<td>No</td>
<td>X</td>
<td>158 535</td>
<td>8</td>
<td>4</td>
<td>300</td>
</tr>
<tr>
<td>E3</td>
<td>No</td>
<td>X</td>
<td>178 681</td>
<td>10</td>
<td>5</td>
<td>502</td>
</tr>
<tr>
<td>E4</td>
<td>No</td>
<td>X</td>
<td>238 506</td>
<td>12</td>
<td>6</td>
<td>998</td>
</tr>
<tr>
<td>E5</td>
<td>No</td>
<td>X</td>
<td>332 324</td>
<td>16</td>
<td>8</td>
<td>1198</td>
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<tr>
<td>E6</td>
<td>No</td>
<td>X</td>
<td>429 635</td>
<td>18</td>
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<tr>
<td>E7</td>
<td>Yes</td>
<td>X</td>
<td>219 805</td>
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<td>156</td>
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<tr>
<td>E8</td>
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<td>558 949</td>
<td>6</td>
<td>3</td>
<td>156</td>
</tr>
</tbody>
</table>

Figure 18. Local mesh refinement of the area around the strain gauge’s position, (a) mesh E4, (b) mesh E5, (c) mesh E6
5.5 Linings, discharger and surrounding components

The linings inside the mill consist of block magnets embedded in hard rubber. During production, iron ore is attracted to the magnetic linings, creating a protective ore layer inside the mill. The linings and the ore layer increase the stiffness of the mill. The discharger attached to the outlet head consists of hard rubber and steel and increases further the stiffness of the mill. Other components which also increase the stiffness are certain parts of the ore flow transition systems attached to the outlet and inlet end of the mill.

These stiffeners must be included in the finite element model to capture the appropriate stiffness of the mill. The components have been meshed and incorporated in the existing model, see Figure 21. The discharger and the linings and ore layer have been modelled as
cylinders with constant thicknesses. In reality, the parts are fixed to the mill with bolts. To model this interaction, completely glued surface-to-surface contact has been used between the meshes.

The exact thickness of the linings and ore layer combination is unknown, and so too is the effective modulus of elasticity. The thickness of the linings decreases with time, due to wear, and the thickness of the ore layer varies during production. The approximate thickness of the discharger is known from drawings, but the effective modulus of elasticity is unknown.

The geometries of the transition system components are given by drawings and the material properties are the same as those for the mill mandrel and heads.

The stiffness of the linings-ore-layer combination and the discharger is a function of the geometry, foremost the thickness, and the effective modulus of elasticity. In order to find the correct effective stiffness of these parts, the model has been solved for constant thicknesses and a varying effective modulus of elasticity, for load cases L1-L6. The thickness of the linings-ore layer has been set to 40 mm for the mandrel and 75 mm for the inlet head. The thickness of the discharger has been set to 375 mm. For each load case and effective modulus of elasticity, the strains have been calculated and compared to the corresponding measured ones in order to find the closest agreement. Since the real thicknesses of the linings-ore-layer and discharger are unknown, the value of the effective modulus of elasticity for these parts can be seen as a stiffness-adjusting parameter and as being unrelated to material properties.

The effective modulus of elasticity has been varied from zero to several times the modulus of steel. Parts of the calculated results for load case L1, L3, L5 and L6 are shown in Table 6. Load case L1 gives the closest agreement with measurements when the stiffness is set to zero. Load case L3, L5 and L6, however, give the best agreements when stiffness is added. Best agreement is given by L5 in combination with S8, followed by L5 in combination with S7 and L6 in combination with S7.
Table 6. Calculated strain ranges for different moduli of elasticity on the linings/orebed and discharger

<table>
<thead>
<tr>
<th>Load case</th>
<th>Case of modulus of elasticity for linings-ore-layer combination and discharger</th>
<th>Modulus of elasticity, % of modulus for mill shell and heads [%]</th>
<th>Deviation in strain range [%]</th>
<th>Circumferential direction</th>
<th>Longitudinal direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>L1</td>
<td>S1</td>
<td>0 0</td>
<td>8.49</td>
<td>282</td>
<td></td>
</tr>
<tr>
<td>L3</td>
<td>S1</td>
<td>0 0</td>
<td>61.9</td>
<td>281</td>
<td></td>
</tr>
<tr>
<td>L3</td>
<td>S2</td>
<td>50 50</td>
<td>25.1</td>
<td>158</td>
<td></td>
</tr>
<tr>
<td>L3</td>
<td>S3</td>
<td>100 100</td>
<td>4.17</td>
<td>101</td>
<td></td>
</tr>
<tr>
<td>L3</td>
<td>S4</td>
<td>120 120</td>
<td>-1.99</td>
<td>85.5</td>
<td></td>
</tr>
<tr>
<td>L3</td>
<td>S5</td>
<td>150 150</td>
<td>-9.66</td>
<td>66.4</td>
<td></td>
</tr>
<tr>
<td>L5</td>
<td>S5</td>
<td>150 150</td>
<td>2.76</td>
<td>88.6</td>
<td></td>
</tr>
<tr>
<td>L5</td>
<td>S6</td>
<td>180 180</td>
<td>-4.19</td>
<td>70.5</td>
<td></td>
</tr>
<tr>
<td>L5</td>
<td>S7</td>
<td>200 200</td>
<td>-8.19</td>
<td>60.9</td>
<td></td>
</tr>
<tr>
<td>L5</td>
<td>S8</td>
<td>210 210</td>
<td>-10.0</td>
<td>56.6</td>
<td></td>
</tr>
<tr>
<td>L6</td>
<td>S5</td>
<td>150 150</td>
<td>0.87</td>
<td>90.7</td>
<td></td>
</tr>
<tr>
<td>L6</td>
<td>S7</td>
<td>200 200</td>
<td>-9.59</td>
<td>63.55</td>
<td></td>
</tr>
</tbody>
</table>

5.6 Applied loads

Gravity forces were prescribed for the entire mill and all the attached components. The calculated normal and friction forces from the charge for each load case were applied to the model. The normal and friction force distributions from the charge for load case L1 and L3 are shown in Figure 22. The tangential, normal and axial forces from the pinion were applied and distributed linearly on the drive wheel over an area of 20 degrees around the area in contact with the pinion, with a maximum at the point of intervention, see Figure 9.

The friction forces from the charge and the tangential forces from the pinion were applied as x and y force-components in the model.

Figure 22. Distribution of normal and friction forces from the charge, max=red and min=blue, (a) load case L1, (b) load case L3

6 Results

In Figure 23 the strains in the longitudinal and circumferential directions around the circumference of the mandrel’s outer surface at the axial position of the strain gauges are
plotted for each load case (L1-L6), with the modulus of elasticity for the linings and discharger set according to S5. The starting point for the plots is shown in Figure 3.

Table 7 shows the maximum strain range for each load case, as well as the obtained reaction forces in the nodes at each bearing support. Table 8 shows the corresponding measured strains and logged reaction forces.

It is found that load case L3 offers the least deviation from the measured values in terms of the strain ranges. In terms of the reaction forces, load case L6 offers the best agreement.

For all the load cases, the obtained reaction force in the node locked in the circumferential direction gives a torque less than 0.03 per mille of the current $M_{\text{mill}}$. This indicates that equation (19) is satisfied for the simulation model.

As can be seen in Table 6 and Table 7, some of the calculations show best agreement in the circumferential strain range, while others show best agreement in the longitudinal strain range and for the reaction forces. In Figure 24 the calculated strains for load case L3 are plotted together with the measured strains. As can be seen in Figure 24, the curves for the calculated and the measured strains are similar in shape and phase.

Figure 25 shows the calculated radial displacements in the mill for load case L3. In order to emphasize the deformation pattern of the mill, the maximum deformation has been scaled to a program default value of 10% of the model size. The remaining deformations are adjusted according to this value.

![Figure 23. Calculated strains plotted for one mill rotation, (a) circumferential strains, (b) longitudinal strains](image)

<table>
<thead>
<tr>
<th>Table 7. Calculated strain ranges and bearing reaction forces</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Load case</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>L1/S5</td>
</tr>
<tr>
<td>L2/S5</td>
</tr>
<tr>
<td>L3/S5</td>
</tr>
<tr>
<td>L4/S5</td>
</tr>
<tr>
<td>L5/S5</td>
</tr>
<tr>
<td>L6/S5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 8. Measured strain ranges and logged bearing reaction forces</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Measured strain range</strong> $\mu \text{mm/mm}$</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>37.98</td>
</tr>
</tbody>
</table>
Discussion

The described MLC software, together with the finite element model, makes it possible to calculate quickly the global displacement field of the entire mill structure for any given process situation. This is achieved by simply typing the present process data and desired charge parameters into the MLC software, calculating the loads and force distributions, exporting them to the finite element model and running the simulation. From the global displacement field the strains, the stresses, the reaction forces, etc. can then be calculated with standard routines for any position of the mill.

The obtained global displacement field of the running mill can be used to evaluate the fatigue performance of the mill and can provide sub-model boundary conditions in detailed calculations aimed, for example, at fracture mechanics analysis or remaining fatigue life assessments.

Present day FEM calculations of large three-dimensional structures are often found to fall short of capturing real structure behaviour. Throughout the modelling process, the
assumptions and estimations have been kept to a minimum to minimize any final deviations between the models and the physical behaviour of the real mill. The agreements between the calculated and the measured strain ranges and reaction forces are considered good for the performed simulations. In the circumferential direction, the strain range difference is for most calculations less than 10%. In the longitudinal direction, the difference is larger, often above 10%, and no single explanation has been found for this. In order to obtain better agreements than have been presented, further refinement of the spatial discretization, the boundary conditions, the material model and the load model might be required. It should be mentioned that potential errors could exist in the strain measurement data due to, for example, misalignment of the gauges from expected directions.

The MLC software shows good agreement with the corresponding hand-calculations, as can be seen in Appendix B and C. The software can in further work be used to investigate in detail how different charge and process parameters influence the mill strains in operation, with the purpose of finding the optimal process parameters, to avoid overloading and permit the safe running of the mill.

Furthermore, the MLC software and finite element model can be used at a design stage for evaluating the stress in the mill and to optimize the mill design for the current loadings. For this purpose the loads can be calculated for mean values of process data over time and peak loads for worst case scenarios.

Mill failures are very expensive due to the high costs of repair and production loss. Consequently, the research described in this paper is providing solutions to problems which the mining industry and mill manufacturers are very eager to solve. The described methodology can be used to evaluate and analyze the strains in mining mills during operation under real-life loading conditions and can be used to improve the mill designs. This will hopefully lead to mills which will show better resistance to the loads which they are subjected to, and to a decreasing number of mill failures and reduced costs.

8 Conclusions

The complexity of modelling the structural behaviour of the mill in operation is high. The agreements between the calculated and the measured values are considered good for the performed simulations and the presented models can be assumed to be accurate enough for engineering purposes.

The results indicate that there is no optimal combination of load case and modulus of elasticity for the linings-ore-layer and discharger. Some combinations show best agreement in the circumferential strain range, while others show best agreement in the longitudinal strain range and for the reaction forces. Load case L6, in combination with an effective modulus of elasticity according to case S5, yields good overall agreement with the strain measurements and logged reaction forces. In the circumferential direction, the strain range difference for this combination is 9.53% and in the longitudinal direction 63.5%.

It has been found that the strains in the mill are highly dependent on the load case, and the correct input of loads into the finite element model is crucial. The final load case and effective modulus of elasticity for the linings and discharger were obtained by comparing the calculated strains and reaction forces to the corresponding measured and logged values. One conclusion is that it is important to have these measurements and logged data for the purpose of comparison, in order to obtain accurate and reliable results. Since the mill operation process is constantly changing over time, it is necessary that the computed results for process data for a certain time period are verified with measurements and logged data for the same time period.

For any constant value of \( \omega, m, P_{\text{motor}} \) and \( V \), it has been found that the normal forces and \( \vec{F} \) increase, while the friction forces and \( \mu \) decrease with decreasing \( R \). The normal forces are the
greatest forces and exert the largest influence on the strains in the mill. Therefore, decreasing \( R \) leads to increasing strains in the mill, even when the friction forces decrease. \( \alpha \) has a large influence on the magnitude of the normal forces. A larger \( \alpha \) displaces the charge to the side, which increases \( h \) and the normal forces.

For any constant values of \( \omega, V, m \) and \( R \), all the forces acting on the mill increase with increasing \( P_{\text{motor}} \). Greater \( P_{\text{motor}} \) increases \( \alpha, M_{\text{friction}}, \mu \) and \( M_{\text{wheel}} \), which all in turn increase the forces acting on the mill. Therefore, the strains in the mill increase with increasing \( P_{\text{motor}} \). An obvious corollary is that the mill stresses decrease with decreasing motor power.

Furthermore, it has been found that the normal forces increase, while the friction forces remain about the same with increasing \( V \) and decreasing \( \rho \) for any constant values of \( \omega, m, P_{\text{motor}} \) and \( R \). One conclusion from this is that the mill stresses decrease with a lowering of \( V \) for a constant \( m \). Therefore, for a constant charge mass, a higher charge density is more advantageous than a lower in order to minimize the stresses in the mill. For this to be valid, it is important that the charge mass does not increase along with the density.

9 Acknowledgements

The author would like to acknowledge thankfully the financial support provided by VINNOVA and LKAB for this paper. Thanks are extended to Dr Johan Tillberg his valuable suggestions and comments for improvement of this article. Further thanks are due to Professor Jan Lundberg for his valuable comments on the work. Finally, special thanks are extended to Professor Uday Kumar for having full faith in me during my research.

10 References


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Appendix A – Charge torque and filling level for the charge with a concave upper surface

The charge with a concave upper surface can geometrically be obtained from a charge with a linear upper surface (Charge A in Figure 1) by subtracting the surface region between the linear surface and the nearest concave surface (Charge B in Figure 1) from the surface region of Charge A. The resulting surface, a crescent, is denoted by Charge C in Figure 1.

This crescent represents the charge with a concave upper surface. The area of Charge C can be written as:

$$A_{\text{charge C}} = A_{\text{charge A}} - A_{\text{charge B}}$$  \hspace{1cm} (A1)

where $A_{\text{charge B}}$ is the area of Charge B and $A_{\text{charge A}}$ is the area of Charge A.

The torque of the charge with a concave upper surface can be written as:

$$M_{\text{charge C}} = d(H, R, \alpha) \cdot m \cdot g$$  \hspace{1cm} (A2)

where $d$ is the horizontal distance between the centre of the mill and the CG of Charge C. $d$ is calculated as:

$$d = S_{\text{charge C}} \cdot \sin(\alpha)$$  \hspace{1cm} (A3)

$S_{\text{charge C}}$ is the radial distance from the mill centre to the CG of Charge C. $S_{\text{charge C}}$ can be calculated from the surface moment equilibrium of the surface regions with the areas $A_{\text{charge A}}$, $A_{\text{charge B}}$ and $A_{\text{charge C}}$. 
\[ S_{\text{chargeC}} = \frac{A_{\text{chargeA}} \cdot S_{\text{chargeA}} - A_{\text{chargeB}} \cdot S_{\text{chargeB}}}{A_{\text{chargeC}}} \]  \hspace{1cm} (A4)

where \( S_{\text{chargeA}} \) and \( S_{\text{chargeB}} \) are the radial distances from the mill centre to the CGs of Charge A and Charge B, respectively. (A4) is the formula for the radial distance to the CG of a crescent with radius \( R \) inside a circle with radius \( r \).

Using Pythagoras’ Theorem and the sine formula, the following relations can be obtained from Figure 1:

\[ R^2 = l^2 + (R - b)^2 \]  \hspace{1cm} (A5)

\[ l = r \cdot \sin(\beta_1/2) \]  \hspace{1cm} (A6)

Here \( b \) is the radial distance, in line with the CGs, between the concave upper line of Charge C and the straight upper line of Charge B. By inserting (A10) into (A6) and (A6) into (A5), and by rewriting (A5), \( b \) can be written as:

\[ b = \frac{-H^2 + 2 \cdot r \cdot H}{2 \cdot R + 2 \cdot H - 2 \cdot r} \]  \hspace{1cm} (A7)

\( H \) is the maximum radial height of the crescent forming the charge with a concave upper surface. In equation (A4):

\[ A_{\text{chargeA}} = \frac{r^2}{2} \cdot (\beta_1 - \sin(\beta_1)) \]  \hspace{1cm} (A8)

and

\[ S_{\text{chargeC}} = \frac{4 \cdot r \cdot \sin^3(\beta_1/2)}{3 \cdot (\beta_1 - \sin(\beta_1))} \]  \hspace{1cm} (A9)

where \( \beta_1 \) is the sector angle of Charge A with respect to the mill centre, calculated as:

\[ \beta_1 = 2 \cdot \arccos\left(\frac{r - (H + b)}{r}\right) \]  \hspace{1cm} (A10)

Further:

\[ A_{\text{chargeB}} = \frac{R^2}{2} \cdot (\beta_2 - \sin(\beta_2)) \]  \hspace{1cm} (A11)

\( S_{\text{chargeB}} \) is the radial distance to the CG of Charge B with respect to the centre of the circle with radius \( R \). \( S_{\text{chargeB}} \) is calculated as:

\[ S_{\text{chargeB}} = \frac{4 \cdot R \cdot \sin^3(\beta_2/2)}{3 \cdot (\beta_2 - \sin(\beta_2))} \]  \hspace{1cm} (A12)

A2
where $\beta_2$ is the sector angle of Charge B with respect to the centre of the circle with radius $R$, given as:

$$\beta_2 = 2 \cdot \arccos\left( \frac{R - b}{R} \right)$$  \hspace{1cm} (A13)

$S_{\text{charge,B}}$ can be calculated as:

$$S_{\text{charge,B}} = S_{\text{charge,Al}} - R \cdot \cos(\beta_2 / 2) + r \cdot \cos(\beta_1 / 2)$$  \hspace{1cm} (A14)

Now, combining the above equations and inserting them into (A2) give $M_{\text{charge}}$ as a function of $m$, $\alpha$, $H$, and $R$ for the charge with a concave upper surface.

The filling level for the charge with a concave upper surface can be written as:

$$V = \frac{A_{\text{charge,C}}}{A_{\text{mill}}}$$  \hspace{1cm} (A15)

Where

$$A_{\text{mill}} = \pi \cdot r^2$$  \hspace{1cm} (A16)

By combining the above equations and inserting them into (A15), $V$ can be written as a function of $H$ and $R$.
Appendix B – Hand-calculation of the normal force acting on one area element for the mill standing still

In order to verify the charge load calculation algorithm in the MLC program, the magnitude of the normal force acting on one area element at a known position on the mill head has been calculated by hand and compared to the corresponding value from the MLC program. The force has been calculated for the conditions prevailing with a turned-off engine, which give an equal axial charge distribution in terms of density and volume and a horizontal linear charge upper surface. The parameters used for the calculation are given in Table 1. Figure 1 shows the charge profile for the load case and a representative illustration of the area element.

Figure 1. Mill charge with area element

Table 1. Data used for the hand-calculation

<table>
<thead>
<tr>
<th>V</th>
<th>P_{motor}</th>
<th>R</th>
<th>α</th>
<th>m</th>
<th>ω</th>
<th>H</th>
<th>ρ</th>
<th>r</th>
</tr>
</thead>
<tbody>
<tr>
<td>[%]</td>
<td>[kW]</td>
<td>[m]</td>
<td>[°]</td>
<td>[tons]</td>
<td>[rpm]</td>
<td>[m]</td>
<td>[kg/m³]</td>
<td>[m]</td>
</tr>
<tr>
<td>40</td>
<td>0</td>
<td>Inf</td>
<td>0</td>
<td>372.1</td>
<td>0</td>
<td>2.704</td>
<td>3570</td>
<td>3.210</td>
</tr>
</tbody>
</table>

The area element has the area, \( A_{element} = 2.684 \times 10^{-4} \text{ m}^2 \), which represents a quadrate whose sides measure 16.38 mm each. The element has the coordinate position, \( x_{element} = 1.0066 \text{ m} \) and \( y_{element} = -1.5091 \text{ m} \). The vertical charge height, \( h_{element} \), over the element is calculated as:

\[
h_{element} = H - r - y_{element} = 1.00277m
\]

The normal force acting on the element is calculated as:

\[
F_{normal \_element} = A_{element} \cdot \rho \cdot g \cdot h_{element} = 9.436091N
\]

The corresponding normal force calculated by the MLC program is obtained as:

\[
F_{normal \_element \_MLC} = 9.432374N
\]

The agreement is good and improves with an increasing number of area elements.
Appendix C – Hand-calculation of the normal force acting on one area element for the mill in operation

In order to verify the charge load calculation algorithm in the MLC program, the magnitude of the normal force acting on one area element at a known position on the mill head has been calculated by hand and compared to the corresponding value from the MLC program. The force has been calculated for the conditions of the mill in operation with an equal axial charge distribution in terms of density and volume and a concave upper charge surface with the radius \( R \). The parameters used for the calculation are given in Table 1. Figure 1 shows the charge profile for the actual load case used for the calculation and a representative illustration of the area element.

The area element has the area, \( A_{element} = 2.001 \times 10^{-4} \text{ m}^2 \), which represents a quadrate whose sides measure 14.14 mm each. The element has the coordinate position, \( x_{element} = 1.8621 \text{ m} \) and \( y_{element} = -1.5004 \text{ m} \), with respect to the mill centre.

In order to obtain the vertical charge height, \( h_{element} \), above the element, the y-value at the charge surface with respect to the mill centre at the \( x_{element} \) position must be calculated. The coordinate position of this point is \( (x_{element}, y_{surface}) \) with respect to the mill centre and \( (X_{surface}, Y_{surface}) \) with respect to the circle with the radius \( R \), see Figure 1 in Appendix A.

The linear distance between the mill centre and the centre of the circle with the radius \( R \) is given as:

\[
d_{between} = R + H - r
\]  

\( X_{surface} \) can be calculated as:

\[
X_{surface} = x_{element} + d_{between} \cdot \sin(\alpha)
\]  

\[
\begin{array}{|c|c|c|c|c|c|c|c|c|} 
\hline
V & P_{motor} & R & \alpha & m & \omega & H & \rho & r \\
[\%] & [\text{kW}] & [\text{m}] & ['] & [\text{tons}] & [\text{rpm}] & [\text{m}] & [\text{kg/m}^3] & [\text{m}] \\
\hline
\hline
\end{array}
\]
and $Y_{\text{surface}}$ can be calculated from $R$ and $X_{\text{surface}}$ by Pythagoras’ equation:

$$Y_{\text{surface}} = -\sqrt{R^2 - X_{\text{surface}}^2} \quad (C3)$$

$Y_{\text{surface}}$ is given by:

$$y_{\text{surface}} = Y_{\text{surface}} + d_{\text{between}} \cdot \cos(\alpha) \quad (C4)$$

and $h_{\text{element}}$ by:

$$h_{\text{element}} = y_{\text{surface}} - y_{\text{element}} \quad (C5)$$

By combining the above equations $h_{\text{element}}$ can be calculated as:

$$h_{\text{element}} = -\sqrt{R^2 - (X_{\text{element}} + (R + H - r) \cdot \sin(\alpha))^2} + (R + H - r) \cdot \cos(\alpha) - y_{\text{element}} = 1.91373 \text{ m} \quad (C6)$$

The normal force on the element can then be calculated as:

$$F_{\text{normal \_element}} = A_{\text{element}} \cdot \rho \cdot g \cdot h_{\text{element}} = 13,424199 \text{ N} \quad (C7)$$

The corresponding normal force calculated by the MLC program is obtained as:

$$F_{\text{normal \_element \_MLC}} = 13,421171 \text{ N} \quad (C8)$$

The agreement is good and improves with an increasing number of area elements.
Appendix D – Calculation of the charge mass from the bearing pressures in the mill in operation

The pressure on the six main hydrostatic bearings is continuously logged during mill operation. The main bearings support the mill in the radial directions. The two bearings positioned at the bottom of the mill are denoted by “Slave”, and the other four bearings, positioned at 30° from the bottom position, are denoted by “Master”, see Figure 1.

![Figure 1. Hydrostatic pressure bearings and bearing forces](image)

The bearing pressure in bar is linearly proportional to the total force, \( F_{bearing} \), in the direction normal to the bearing surface according to:

\[
P_{bearing} = 4.3127 \cdot 10^{-5} \cdot F_{bearing} + 0.1240 \quad \text{(D1)}
\]

The total normal force on each bearing can then be calculated as a function of the bearing pressure in bar as:

\[
F_{bearing} = \frac{P_{bearing} - 0.1240}{4.3127 \cdot 10^{-5}} \quad \text{(D2)}
\]

During mill operation, the tangential and normal directed forces on the drive wheel, \( F_{wheel} \) and \( F_{wheel normal} \), creates vertical upward lifting forces on the mill. These vertical upward lifting forces reduce the pressure on the bearings during operation and must be included in the calculations of the charge mass on the running mill.
The pinion is in contact with the drive wheel 20° below the horizontal line, see Figure 1. The vertical y-component of $F_{\text{wheel}}$ can then be calculated as:

$$F_{\text{wheel}\_y} = F_{\text{wheel}} \cdot \cos(20°) \quad (D3)$$

and the vertical y-component of $F_{\text{wheel\_normal}}$ can be calculated as:

$$F_{\text{wheel\_normal}\_y} = F_{\text{wheel\_normal}} \cdot \sin(20°) \quad (D4)$$

$F_{\text{wheel}}$ and $F_{\text{wheel\_normal}}$ are calculated in Appendix E.

The total equivalent charge mass, $m$, is then calculated by the sum of the vertical force components on the bearings, the addition of the vertical uplifting forces and the subtraction of the empty mill mass and attached components:

$$m = \cos(30°) \sum F_{\text{bearing\_master}} + \sum F_{\text{bearing\_slave}} + F_{\text{wheel\_y}} + F_{\text{wheel\_normal\_y}} - m_{\text{mill\_empty}} \quad (D5)$$

In equation D3, $m_{\text{mill\_empty}}$ is the total mass of the empty mill and attached components, e.g. the trommel sieve, linings and discharger. $g$ is the acceleration due to gravity.
**Appendix E – Calculation of the drive wheel forces**

In this appendix the drive wheel forces in the tangential direction \( F_{\text{wheel}} \), the axial direction \( F_{\text{wheel axial}} \) and the normal direction \( F_{\text{wheel normal}} \) are calculated, see Figure 1. The forces are calculated from the torque on the drive wheel \( M_{\text{wheel}} \) and the gear data of the drive wheel and the pinion. \( M_{\text{wheel}} \) is obtained from equation (19). The gear data used for the calculations are found in Table 1. For more information see SMS Handbook 515:1987 (1987).

![Figure 1. Drive wheel forces, a) drive wheel and pinion seen from the outlet end, b) drive wheel seen from the long side](image)

<table>
<thead>
<tr>
<th>Addendum modification factors</th>
<th>Number of teeth</th>
<th>Pressure angle [°]</th>
<th>Helix angle [°]</th>
<th>Real module [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drive wheel ( x_1 )</td>
<td>Pinion ( x_2 )</td>
<td>Drive wheel ( z_1 )</td>
<td>Pinion ( z_2 )</td>
<td>( x_{\text{sa}} )</td>
</tr>
<tr>
<td>0</td>
<td>0.281997</td>
<td>220</td>
<td>23</td>
<td>20</td>
</tr>
</tbody>
</table>

The pitch diameter is calculated as:

\[
D_{\text{pitch}} = \frac{\lambda \cdot z_1}{\cos(\xi_{\mu})} \tag{E1}
\]

and the angle of action at the pitch circle in the transverse section is calculated by:

\[
\xi = \arctan \left( \frac{\tan(\xi_{\mu})}{\cos(\xi_{\mu})} \right) \tag{E2}
\]

where \( \xi \) must be in radians. The basic circle diameter is calculated by:

\[
D_{\text{basic}} = D_{\text{pitch}} \cdot \cos(\xi) \tag{E3}
\]

and the basic circle radius by:
The involute of the angle $\xi_u$ is calculated as:

$$\text{inv}(\xi_u) = \tan(\xi) - \xi + \frac{2 \cdot (x_1 + x_2) \cdot \tan(\xi_u)}{z_1 + z_2}$$  \hspace{1cm} (E5)

From literature table data $\xi_u$ is obtained as:

$$\xi_u = 20.4667^\circ$$  \hspace{1cm} (E6)

Now the operating radius of the drive wheel can be calculated as:

$$r_{\text{wheel}} = \frac{r_{\text{base}}}{\cos(\xi_u)}$$  \hspace{1cm} (E7)

$r_{\text{wheel}}$ is the radius for the force intervention on the drive wheel. The engagement force on the drive wheel is calculated by:

$$F_i = \frac{M_{\text{wheel}}}{r_{\text{base}}}$$  \hspace{1cm} (E8)

This gives the tangential force on the drive wheel as:

$$F_{\text{wheel}} = F_i \cdot \cos(\xi_u)$$  \hspace{1cm} (E9)

and the normal force on the drive wheel as:

$$F_{\text{wheel, normal}} = F_i \cdot \sin(\xi_u)$$  \hspace{1cm} (E10)

The axial force on the drive wheel is calculated by:

$$F_{\text{wheel, axial}} = F_i \cdot \tan(\xi_u)$$  \hspace{1cm} (E11)
Paper B

Structural analysis of a rotating mining mill with the finite element method

Structural analysis of a rotating mining mill with the finite element method

Filip Berglund

Division of Operation and Maintenance Engineering, Luleå University of Technology, 971 87 Luleå, Sweden, filip.berglund@ltu.se, +46-(0)920-49 38 20

Abstract

In this paper, a structural analysis, using the finite element method, of a rotating grinding mill and the results from wireless strain measurements on a mill in operation, are presented. Stresses, strains and displacements in the mill structure have been calculated with a numerical quasi-static simulation model of a running mill. In order to verify the computed results, circumferential and longitudinal strains have been measured on the outer mill shell surface. The registered strain range for one complete rotation of the mill has been compared with the corresponding calculated strain range. As the difference is less than 10%, it is assumed that the computational model is accurate enough for engineering purposes.

Because of the dynamic loads inside the rotating mill, the most common failure type is fatigue cracking. Due to this, critical areas for fatigue have been identified and fatigue verified according to the standard procedures described in BS 7608:1993 (Code of practice for fatigue design and assessment of steel structures). It is found, that the welded joints between the shell and the end plates in the mill do not fulfil the requirements for infinite fatigue life.

Keywords: Rotating grinding mill; Finite element analysis; Wireless strain measurement; Fatigue; Mill charge analysis; Mill load analysis

1 Introduction

Rotating grinding mills usually work all around the clock, under heavy and fluctuating loads. Because of the severe working conditions, the most common failure type is fatigue [2]. Due to increasing production demands the mills are today run with greater loads than when first installed. After many years in service, fatigue cracks and associated failures have recently begun to appear at an increasing rate in certain mills. In order to locate areas critical for fatigue and to evaluate the fatigue strength of critical details, a structural analysis based on current loading is highly needed. A structural analysis of this kind for a rotary kiln has recently been reported [6], but similar works on mills are scarce.

The structural analysis in this paper concerns the second of two pebble mills in a production line situated in northern Sweden. The analysis has been performed with the finite element method [11] and the numerical results have been verified by in situ strain measurements on the mill during operation.
As a first step in the analysis, the charge behaviour was studied in detail in order to determine the shape of the charge in a running mill and the loads acting on the mill. A mill model was designed and meshed with 3D (three-dimensional) solid elements. After imposing real structure constraints and loading, a quasi-static and linear elastic small deformation calculation of the displacement field was performed. From the displacement field structural stresses and strains were obtained with standard routines. Finally, fatigue verification was performed of critical welded joints in the mill. The computed maximal principal stress range in the welds was compared to the corresponding fatigue limits obtained from standard code of practice BS 7608:1993 [3].

2 Description of the physical mill

The mill under investigation has a diameter of 6.5 m and a length of 8.5 m, and the average mill shell thickness is 40 mm. The rotation speed of the mill is fairly constant at 12.75 rpm (revolutions per minute), which is approximately 75% of the critical speed. The iron ore material enters the mill at the inlet end and is crushed and ground to a fine powder inside the mill, in a process where larger ore lumps grind smaller ones. At the outlet end ore powder is lifted out of the mill and poured into a trommel sieve. The finest powder grades pass the trommel and proceed to further enrichment and processing.

The mill is, in plain terms, a rotating cylindrical barrel with three main parts: the shell, the inlet head and the outlet head. The heads are the circular plates attached to each end of the mill, see Figure 6. The inside surface of the mill is covered with magnetic linings. The mill is equipped with two steering wheels, one at each end, supported by three hydrostatic pressure bearings preventing radial displacement of the mill. Axial movement of the mill is prevented by two vertical pressure bearings at the outlet end, one for each direction. The pressures on the main load supports are continuously logged.

The mill is driven by a single electrical motor, via a pinion, acting on a tooth drive wheel attached to the mill shell. The installed motor power is 4500 kW. The material in the mill is ordinary construction steel with the properties shown in Table 1.

<table>
<thead>
<tr>
<th>Property</th>
<th>Shell, inlet and outlet head</th>
<th>Drive wheel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module of elasticity [MPa]</td>
<td>210 000</td>
<td>167 000</td>
</tr>
<tr>
<td>Poisson’s ratio [-]</td>
<td>0.29</td>
<td>0.29</td>
</tr>
<tr>
<td>Yield strength [MPa]</td>
<td>220</td>
<td>220</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>7800</td>
<td>7200</td>
</tr>
</tbody>
</table>


3 Charge and loading analysis

In the structural analysis the loads acting on the running mill are needed. In order to determine the loads due to the charge, the distribution of the charge inside the running mill must be estimated, as well as the total charge mass.

3.1 Charge analysis

The charge consists of ore lumps of varying size, typically from 35 mm in diameter and downward. The charge also contains a large amount of water. In operation the axial cross-section upper surface of the charge is typically concave [1], [4], [5], [7] and [12]. To simplify calculations the upper surface has been assumed to be straight. It is further assumed that the charge is uniformly distributed along the mill. The position of the charge circumferentially is obtained from the power equilibrium of the mill and motor. During operation the motor rotates the charge to one side, see Figure 3. The displacement of the charge, from zero torque position, is a function of the motor power, the mill rotation speed, the total charge weight and the filling level inside the mill.
The power equilibrium equation of the mill and motor can be written as:

\[ P_{\text{mill}} = P_{\text{motor}} \]  

(1)

where \( P_{\text{mill}} \) is the power of the mill and \( P_{\text{motor}} \) is the motor power. The mill power can be written as:

\[ P_{\text{mill}} = M \cdot \omega \]  

(2)

In the equation, \( M \) is the torque of the charge and \( \omega \) is the angular speed of the mill. From Equation 1 (Eq1) and Eq2, the torque of the charge can be written as:

\[ M = \frac{P_{\text{motor}}}{\omega} \]  

(3)

The charge torque can also be calculated as:

\[ M = d(H, \alpha) \cdot m \cdot g = \frac{4 \cdot R \cdot \left(1 - \frac{(R - H)^2}{R^2}\right)^{\frac{3}{2}}}{6 \cdot \text{arccos} \left(\frac{R - H}{R}\right) - 3 \cdot \sin \left(2 \cdot \text{arccos} \left(\frac{R - H}{R}\right)\right) \cdot m \cdot g} \]  

(4)

In the equation \( d \) is the horizontal distance between the mill centre and the centre of gravity (CG) of the charge, \( m \) is the total charge mass and \( g \) the acceleration due to gravity, which is set to 9.82 m/s². \( d \) is a function of the charge height \( H \) and the lifting angle of the charge \( \alpha \), see Figure 3. \( R \) is the inner radius of the mill. \( H \) is directly related to the filling level \( V \), which is defined as the ratio between the charge volume and the total available volume inside the mill, see Eq5:

\[ V = \frac{\text{arccos} \left(\frac{R - H}{R}\right) - \frac{1}{2} \cdot \sin \left(2 \cdot \text{arccos} \left(\frac{R - H}{R}\right)\right)}{\pi} \]  

(5)

The charge inside the mill is undergoing dynamic changes [9]. Parameters like the ore density, the amount of water in the charge, the filling level and the motor power all vary under production. To be able to compare computed and measured strains, the charge profile needs first to be determined for the specific period during the strain measurement.

A total charge mass of 335.8 tons was calculated from the bearing pressures by subtracting the mass of the empty mill and installed components from the total mass logged on the bearings at the specific time. For the same period, a motor power of 4152 kW was obtained from logged data.

In order to understand how the charge torque is related to the filling level and lifting angle for the obtained charge mass, the charge torque was calculated with Eq4 and Eq5, for filling levels from 0-100% and lifting angles from 0-90°, see Figure 3. For a given charge mass, the maximal charge torque is obtained at a lifting angle of 90° and a minimum filling level. An increasing filling level leads to a decreasing
charge torque, and an increasing lifting angle, up to 90°, leads to an increasing charge torque.

![Figure 3. Relation between charge torque, filling level and lifting angle for a constant charge mass of 335.8 tons: (a) 3D plot, (b) cross-section at α=31.9°, (c) cross-section at V=35%](image)

In order to describe the charge distribution geometrically, the filling level and lifting angle need to be obtained. From the logged motor power and the mill rotation speed, the charge torque \( M_{\text{used}} \) was calculated with Eq3. When the charge torque is known, only the combinations of the filling level and lifting angle which give the calculated charge torque need to be treated. The filling level and lifting angle are both unknown parameters and the one needs to be estimated in order to calculate the other one.

The filling level was chosen for estimation, because it is more suitable to estimate than the lifting angle. When the total charge mass is known, the filling level is directly related to the charge density \( \rho \), see Eq6 below. \( L \) is the inner length of the mill.

\[
V = \frac{m}{L \cdot \pi \cdot R^2 \cdot \rho}
\]  

(6)

In order to estimate the filling level, the charge density was calculated. The charge consists of crushed and water-filled magnetite and based on this knowledge the charge density was calculated by the following equation:

\[
\rho \approx \rho_{\text{magnetite solid}} \cdot \gamma_{\text{crush}} \cdot \gamma_{\text{water}} = 3600 \text{ kg/m}^3
\]  

(7)
In the equation, $\rho_{\text{magnetite solid}}$ is the density of solid magnetite, which is from 4900-5200 kg/m$^3$. Due to the fact that a certain amount of the charge is gangue minerals, a lower value of 5000 kg/m$^3$ was used for the calculation. $\gamma_{\text{crush}}$ is a reducing factor for the crushing with a value of 0.6 and $\gamma_{\text{water}}$ is an adding factor for the water with a value of 1.2. The factors were obtained from printed tables where values for solid, crushed and water-filled densities are given for a number of rock types [13]-[17]. The calculated charge density was verified with the mining company and is considered to be within the limits for normal production.

According to the mill specifications, the filling level should be in the range of 30-40%. To be able to estimate a suitable filling level, the charge densities and lifting angles were calculated and investigated for a number of filling levels, see Table 2. The corresponding charge profiles for the different combinations were visualized graphically and compared with each other for the most life-like look. The charge profiles for the filling levels of 30%, 35% and 40% are displayed in Figure 4. All the combinations in Table 2 give the charge torque of $M_{\text{load}}$.

Table 2. Combinations of filling level, lifting angle and charge density for a constant charge mass of 335.8 tons

<table>
<thead>
<tr>
<th>Charge profile [nr]</th>
<th>Filling level [%]</th>
<th>Lifting angle [°]</th>
<th>Density [kg/m$^3$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>27.5</td>
<td>28.02</td>
<td>4685</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>29.23</td>
<td>4295</td>
</tr>
<tr>
<td>3</td>
<td>32.5</td>
<td>30.53</td>
<td>3969</td>
</tr>
<tr>
<td>4</td>
<td>35</td>
<td>31.93</td>
<td>3681</td>
</tr>
<tr>
<td>5</td>
<td>37.5</td>
<td>33.46</td>
<td>3436</td>
</tr>
<tr>
<td>6</td>
<td>40</td>
<td>35.12</td>
<td>3221</td>
</tr>
<tr>
<td>7</td>
<td>42.5</td>
<td>36.95</td>
<td>3032</td>
</tr>
<tr>
<td>8</td>
<td>45</td>
<td>38.98</td>
<td>2863</td>
</tr>
</tbody>
</table>

Figure 4. Charge profiles with the CG marked for different filling levels: (a) $V=30\%$, (b) $V=35\%$, (c) $V=40\%$

A filling level of 35% gives a density close to the one calculated in Eq7. This filling level is also in the middle of the 30-40% range. On the basis of this and the charge profile comparison, charge profile nr 4 was chosen to be used for the load assessment on the mill.
3.2 Loading analysis

In order to obtain the loads from the charge, the charge is assumed to act like a fluid with friction between the charge and the mill. Based on this assumption, the loads acting on the mill are the following:

- Normal forces $F_{\text{normal}}$ from the charge, acting on the mill shell and heads
- Friction forces $F_{\text{friction}}$ from the charge, acting on the mill shell and heads
- Tangential forces $F_{\text{wheel}}$ from the pinion, acting on the drive wheel. Because of bevelled gearings, there are also force components in the axial mill direction $F_{\text{wheel axial}}$
- The self-weight of the mill and attached components caused by gravity
- Radial and axial reaction forces at bearing surfaces, $F_{\text{bearing}}$ and $F_{\text{bearing axial}}$

Figure 5 shows simplified free body diagrams of the forces acting on the mill during operation. The green arrows represent the forces acting on the mill heads and the red arrows the forces acting on the shell. The friction between the mill and the hydrostatic bearings is neglected.

![Free body diagrams of the mill](image)

Figure 5. Free body diagrams of the mill: (a) view from long side, (b) cross-section at the outlet head, (c) cross-section at the middle of the mill

During operation, the following torque equilibrium condition exists for the mill:

$$M = M_{\text{friction}} = M_{\text{wheel}}$$  \hspace{1cm} (8)
where $M_{\text{friction}}$ is the total torque from the friction forces and $M_{\text{wheel}}$ is the total torque from the tangential forces acting on the drive wheel.

From the obtained charge profile, $m$, $M_{\text{used}}$, and Eq8, the size and distribution of the forces acting on the mill, as well as the coefficient of friction between the charge and the mill $\mu$, were calculated. Table 3 shows the direction independent summation of the normal, friction and drive wheel forces acting on the mill.

<table>
<thead>
<tr>
<th>$F_{\text{normal}}$ [kN]</th>
<th>$F_{\text{friction}}$ ($\mu=0.189$) [kN]</th>
<th>$F_{\text{wheel}}$ [kN]</th>
<th>$F_{\text{wheel axial}}$ [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5241</td>
<td>991</td>
<td>701</td>
<td>86</td>
</tr>
</tbody>
</table>

### 4 Finite element model

From the physical model a finite element model was created with material, constraints and loads implemented in a way to simulate the real conditions of the mill during operation, see Figure 6. Because of the uneven loading conditions, no symmetry assumptions could be made and a full model analysis was required.

The mill was meshed with 8 node 3D solid elements. A total of 539 415 nodes was used for the entire mesh. The geometry was simplified to include only dimensions which were relevant to the structural strength. Bolt holes and small corner radii were removed from the model. Mesh refinement was performed of corner radii on locations critical for fatigue, see Figure 7. In general, 2-3 elements were used across the thickness of the mill, except in the thinnest areas, where only 1 element was used. The trommel was modelled as a 0D point mass. The point mass was connected to the mill outlet using 1D elements.

The gravity was set to the whole mill and all the attached components. The total normal force and friction force in Table 3 were distributed and applied in the model. The forces from the pinion were distributed over an area on the drive wheel with a maximum at the point of intervention. The nodes at the place for the main bearings were constrained in the radial direction. The nodes at the position of one of the axial bearings were constrained in the axial direction. One node was constrained in the rotational direction to prevent singularity of the model during calculation. The performed analysis was static and linear. Non-linear analysis was considered unnecessary because of the small deformations.
5 Results and verifications

The results from the finite element calculation indicate that no stresses in the mill are near the yield strength for the material. Figure 8 shows the Von Mises stresses in the most loaded half of the mill. In Figure 8, a vertical plane has been used to divide the mill into two sections in order to display the inner surface of the mill. The highest stress of 60.1 MPa is located at the outer welded joint connecting the inlet head to the shell. The lowest stress is obtained at an outer position of the drive wheel.
Figure 9 shows the radial displacements in the mill, with the maximum deformation scaled to 10% of the model size. The largest radial displacement is located near the middle of the mill. Figure 10 shows the stresses in the circumferential and longitudinal directions, plotted for one mill rotation at the axial mill position for the maximal radial displacement. The start point for the plots is given in Figure 3.

The obtained reaction force in the node locked in the circumferential direction gives a torque close to zero. This indicates that Eq8 is satisfied for the simulation model.

Figure 11 shows the maximum and minimum principal stresses in the mill. These have been used for the fatigue strength verification of the welded joints in the mill.
5.1 Verification with strain measurements

In order to verify the results from the structural analysis, strain measurements were taken on the mill during operation. Two strain gauges were placed close to each other on the outer mill shell, at a position between the outlet end and the drive wheel. The exact position of the strain gauges was registered in order to allow comparison with FEM-computed results for the same position. One strain gauge was placed in the circumferential direction and the other one was placed in the longitudinal direction. The measurements were performed with wireless strain gauge equipment.

The strains were measured during several mill rotations and the results from this are shown in Figure 12. The measured strains for one mill rotation were taken out and compared with the corresponding calculated strains; see Figure 12b and Figure 13. The start point for the plots in Figure 13 is according to Figure 3. In the circumferential direction, the calculated strain range is 50.5 micro strains and the measured one is 46.6 micro strains. In the longitudinal direction, the calculated strain range is 28.2 micro strains and the measured one is 23.9 micro strains. When comparing these results, only the strain range should be considered. This is because the strains obtained from the finite element analysis show the actual strain in the structure. The strain levels from the measurement are dependent on the calibration and the already existing strain in the mill when the strain gauges were fixed. The
comparison shows that the calculated circumferential strain range is 8.3% higher than the measured one. The same comparison for the longitudinal direction shows that the computed strain range is 18.3% higher than the measured one.

![Figure 12](image1.png)

**Figure 12.** Measured strains in the circumferential and longitudinal direction: (a) curves for several mill rotations, (b) curves for one mill rotation

![Figure 13](image2.png)

**Figure 13.** Calculated strains plotted for one mill rotation at the axial mill position for the strain gauges' location

### 5.2 Fatigue strength verification

The critical areas for fatigue in the mill are the welded joints [10]. The locations for these are marked in the cross-section view of the mill shown in Figure 14. Each weld goes along the complete circumference of the mill. The fatigue strength of the welds has been verified with BS 7608:1993 (Code of practice for fatigue design and assessment of steel structures) [3]. This code was used during the design and manufacturing of the mill.

In the present research work, the code has been used to categorize the welded joints into different classes depending on the type, geometry and stress direction (see Table 4). Each class has a corresponding fatigue limit. The fatigue limit for each weld has been compared with the calculated maximal principal stress range in the weld. Joints
where the maximal principal stress range is under the fatigue limit are considered to have infinite fatigue life. In the joints where the fatigue limit is exceeded, failure will sooner or later occur.

Because of the residual stresses in the welded areas, both negative and positive stresses must be used when calculating the stress range [8]. For this reason, the Von Mises stresses cannot be used, because the Von Mises equation only delivers positive values for all the stresses. Instead, the largest maximum and minimum principal stresses must be calculated and the stress range obtained from these.

One mill rotation is equivalent to one fatigue cycle and because of the mill rotation each position in the cross-section shown in Figure 14 undergoes all the stress variations along the circumference for the specific radial and axial location. In Table 4, the largest maximum and minimum principal stresses, as well as the maximal principal stress range, have been calculated for the welded joints in Figure 14.

![Figure 14. Locations for the welded joints in the mill](image)

Table 4. Calculated principal stress ranges for the welded joints in the mill

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
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<td>-67.2</td>
<td>69.4</td>
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</tr>
<tr>
<td>2</td>
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<td>-11.7</td>
<td>46.7</td>
<td>F</td>
<td>41</td>
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</tr>
<tr>
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<td>-11.2</td>
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<td>-9.6</td>
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<td>18.8</td>
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<td>53</td>
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</tr>
</tbody>
</table>

6 Discussion and conclusions

The curves for the plotted stresses in the circumferential and longitudinal directions at the investigated cross-section for the maximal radial displacement shown in Figure 10 are similar. For one mill rotation, the curves show negative stress for the upper portion of the mill and positive stress for the lower portion of the mill. During the rotation of the mill, each part of the cross-section undergoes alternating compression and tension. This clearly illustrates the alternating loading mechanism leading to fatigue in the mill material.
The agreement between the calculated and the measured strain range is good both in the circumferential and the longitudinal direction. In the circumferential direction, the difference is less than 10%. Based on this, the computational model is considered to be accurate enough for engineering purposes. It is also believed that the global displacement field of the entire mill structure is accurate enough to provide realistic sub-model boundary conditions in detailed calculations, with fracture mechanics, of the remaining useful life of mill parts subjected to fatigue cracks.

The performed fatigue assessment of the mill indicates that the calculated stress ranges in the welded joints at position 1, 2, 15 and 16 exceed the fatigue limits for the particular weld types. The conclusion is made that the connecting welds between the heads and the shell cannot be considered to have an infinite fatigue life. This is in agreement with the failure history of the particular mill in question, where almost all the fatigue crack failures have occurred in these regions. Remarkable is that the fatigue limits at position 1 and 16 are heavily exceeded and this is probably the reason for the frequent failures of the mill in these areas. A stress range just over the fatigue limit would not necessarily lead to failures, but as the limit is so heavily exceeded, failures will occur.

At the time when the mill was built, advanced computer-aided design and finite element methods were not applicable and the stresses could not fully be estimated in the mill. This could be the reason for the undersized connections between the mill heads and the shell.

Mill failures are very expensive due to the high costs of repair and production loss. Because of this, the work described in this paper is highly coveted by the mining industry and the mill manufacturer. The described methodology can be used to evaluate the fatigue strength in the mills and to improve the mill designs to resist better the loads which they are subjected to. This will lead to a decreasing number of mill failures and reduced costs.

7 Acknowledgements

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8 References


Paper C

Health monitoring of mining mills with infrared thermography

Health monitoring of mining mills with infrared thermography

Filip BERGLUND and Aditya PARIDA
Division of Operation and Maintenance Engineering, Luleå University of Technology,
971 87 Luleå, Sweden, filip.berglund@ltu.se

ABSTRACT

Infrared thermography has been investigated as a non-destructive testing method for health monitoring of rotating mining mills. The idea is to monitor the thermal differences on the mill surface with a thermal camera in order to detect and monitor cracks and other material damage in the mill material. A general description of the thermography technique, the theory behind it, the advantages and drawbacks, as well as its use in other applications, is presented. To test the usefulness of the method, real life measurements with an IR-camera have been performed on several mining mills at a mining company located in northern Sweden. The results from these measurements are presented and discussed in this article. Infrared thermography is a fast and relatively cheap inspection method which can find and monitor material damage in rotating mining mill and help the maintenance personnel in taking corrective maintenance decisions.

1. INTRODUCTION

Historically, thermography has been used by the military to observe enemy movement during the night and also to monitor the temperature change in organs and tissues of patients in hospitals (Meola et al., 2004). Then thermography found its application in industry when the relationship between mechanical properties and temperature was better understood and clarified, and as the technology advanced. The applications of the IR-camera started in a small scale in the 1970’s, in the aviation industry. Today, thermography has developed into an important NDE (Non-Destructive Evaluation) method for industry with wide applications. Apart from condition inspections of industrial machines, the method is frequently used for inspection of overheating in electric boards (Jeong et al., 2010) and heat leakage from poor connections in high voltage lines, buildings and pipelines used for district heating (Nissen et al., 2010).

In earlier work the authors performed an extensive literature study of different NDT (Non-Destructive Testing) condition monitoring methods suitable for the mining industry (Nordström et al., 2009). Out of many methods, a few were selected as suitable for health monitoring of rotating mining mills. One of the suitable methods found was infrared thermography. With this background, the authors have used an infrared (IR) image camera to monitor the surface temperature of several mining mills during their operation. The thermal imaging was undertaken in order to find cracks and other material damage in the mills. The mills are located at a mining company located in northern Sweden. The mills investigated are both autogenous (AG) and semi-autogenous (SAG) and of different sizes depending on the types. These mills have in general a diameter of 3.9 metres (m) and 6.5 m and are 8 m long. The raw material from the mine is crushed into smaller pieces and brought into the concentration plant as raw material for the mining mills. The mills grind the iron ore to an optimum size, before it can be further processed. The mining mills operate under a constant speed and are driven by one to two electric motors.

2. TECHNICAL BACKGROUND

An infrared thermography set-up often includes a camera with changeable lenses and a computer for complex analysis. The core of the camera is the infrared detector, which absorbs the IR-energy emitted by the object and converts it into electrical voltage. It is possible to obtain a 2-dimensional (2D) image by
using different prism set-ups in the camera and thereby have the possibility of measuring in both the vertical and the horizontal directions. The IR camera can monitor temperatures in the range from -20°C up to 1500°C and above, depending on the equipment. IR thermography can be divided into two approaches, the passive approach and the active approach. The passive approach tests materials and structures which are naturally at a different temperature, sometimes higher, than the ambient temperature. In the case of the active approach, an external stimulus is necessary to obtain relevant thermal contrasts. The positive aspects of thermography are as follows: a fast inspection rate, non-contact measurement, remote applications, secure measurement, quite an easy interpretation, relatively low equipment costs and usefulness for a wide span of applications. The drawbacks of this method are as follows: poor capability of detecting dislocations which do not create sufficient changes in the thermal field, emissive problems, and possible difficulty in detecting deeper defects than surface dislocations and flaws (Maldague, 2001; Kaplan, 1999).

Belgon (1956) developed infrared radiometric techniques for detecting temperature changes. This method could calculate the temperature changes associated with stress by measuring the infrared radiation emitted from the surface of a solid material. However, until 1980, thermography could not be developed fully as an NDE method due to the sensitivity limitation of the instrumentation. With the development of electro-optics and signal processing, an advanced infrared imaging camera (IR Camera) appeared for use. This camera could have a temperature resolution of 10⁻⁰⁰°C and a spatial resolution up to several μm, and could make quantitative stress-strain analyses based on temperature which later on were developed for monitoring mechanical damage.

Infrared thermography is a suitable method for producing thermal images from invisible radiant energy emitted from stationary or moving objects at any distance and without surface contact. The thermovision camera can be used to measure the temperature rise prior to a fatigue crack. Due to thermo-mechanical coupling, infrared thermography provides a non-destructive, non-contact and real time test to observe the physical process of metal degradation and to detect intrinsic dissipation (Yang, 2003). Therefore, infrared thermography can measure material damage and rapidly evaluate the limit of progressive damage under load. Infrared thermography is an attractive and relatively unexplored method for the NDE of the fatigue limits of materials and structures. For surface crack-length measurement, laser pulse heating and thermal microscopy have emerged as new techniques. In this case, cracks can be detected by the increased thermal radiance emitted by an open crack cavity or due to the effect of the crack’s thermal barrier behaviour on the surface temperature where a thermal gradient exists. The increased cavity radiance can be detected in a simple and effective manner for cracks which are open at least to 3 to 5 μm (micrometres).

3. THEORY

Infrared energy is emitted from all materials which have a temperature above 0 degrees Kelvin (~273°C). Infrared energy is a form of electromagnetic radiation. Electromagnetic radiation is emitted energy in the form of waves which travel at the speed of light. The shorter the wave length is, the more energy is carried by the rays. The wavelength (λ) of the infrared radiation ‘band’ is 0.78 to 1000 μm. This is longer than the wavelength of visible light, yet shorter than radio waves (Young et al., 2004). The wavelengths of infrared radiation are categorized into three sub-bands: the near infrared (λ = 0.78 - 1.5 μm), the intermediate infrared (λ = 1.5 - 5.6 μm) and the far infrared (λ = 5.6 - 1000 μm) (NEC San-ei Instruments, 2006). The infrared energy radiated from an object comes from the movement of atoms and molecules in the surface of the object. Higher temperature means more movements in the molecules and atoms, which lead to greater intensity of the emitted infrared energy. As well as emitting infrared energy, objects are also reflecting, absorbing and transmitting infrared energy. When an object has the same temperature as its surroundings, the amount of emitted energy is equal to the amount of absorbed energy.

The emissivity, $e$, of an object is defined by Equation 1 (Eq 1) (Young et al, 2004):

$$e = \frac{W_e}{W_{em}}$$  \hspace{1cm} (1)

where

$W_e$...
\[ W_o = \text{total radiant energy emitted by a body at a given temperature } T \]
\[ W_{bb} = \text{total radiant energy emitted by a blackbody at a given temperature } T \]

A blackbody is a theoretical surface which absorbs and re-radiates all the IR-energy that it receives without reflecting or transmitting any IR energy, see Eq 2.

\[ \text{absorptivity} = \text{emissivity} = 1 \]  

A perfect blackbody does not exist in nature, but a real life object can be described by Eq 3.

\[ \text{absorptivity} + \text{transmissivity} + \text{reflectivity} = 1 \]  

It is possible to create a blackbody-type source by creating a small hole where the ratio between the length (L) and radius (r) is bigger than or equal to 6, see Eq 4 and FIGURE 1. The blackbody-type source is the empty space of the hole.

\[ \frac{L}{r} \geq 6 \]  

![FIGURE 1. Blackbody-type source](image)

The energy radiated from a blackbody can be described by Planck’s Law (Planck, 1914), see Eq 5.

\[ W_x = \frac{C_1}{\lambda^4 \left( e^{C_2/\lambda T} - 1 \right)} \]  

where,

\[ W_x = \text{Spectral radiant emittance per unit wavelength and unit area } [W/cm^2 \cdot \mu m] \]
\[ C_1 = \text{First radiation constant} = 3.7418 \cdot 10^{16}[W/cm^2 \cdot \mu m] \]
\[ C_2 = \text{Second radiation constant} = 1.4388 \cdot 10^{8} [\mu m \cdot K] \]
\[ \lambda = \text{Wavelength } [\mu m] \]
\[ e = \text{Emissivity} \]
\[ T = \text{Absolute temperature } [K] \]

In order to obtain the total emitted energy radiated from a blackbody, Eq 5 must be integrated through all the wavelengths (0 to infinity). The resulting equation is known as the Stefan-Boltzmann equation (Boltzmann, 1884), see Eq 6.

\[ W = e \sigma T^4 \]  

where,

\[ \sigma = \text{Stefan-Boltzmann constant} = 1.4388 \cdot 10^8 [\mu m \cdot K] \]

From Eq 6 the temperature of an object surface can then be calculated by Eq 7.
\[ T = \left( \frac{W}{e\sigma} \right)^{1/4} \] (7)

During measurement with an IR-camera, the energy \( W \) emitted from an object is monitored and converted into an electrical signal by the imaging sensor (microbolometer) in the camera and the temperature \( T \) is displayed on a monitor as a colour or monochrome thermal image.

3.1 DETERMINING EMISSIVITY

To obtain the correct temperature \( T \) of an object, when measuring with an IR-camera, it is crucial to determine the correct emissivity \( e \) value of the object surface, see the relation between the surface temperature and the emissivity in Eq 7.

The emissivity of an object can be obtained in many ways, two of which will be described below:

1) By using a printed table
   Many books and other works of literature contain tables of emissivity for various materials. However, in order to use tabled values, the object surface must be identical to one in the table. Otherwise, the tabled values can only be used as references.

2) By comparison of the object surface with the surface of a blackbody
   a) Stabilize the temperature of the object.
   b) Make a small hole in the object surface according to FIGURE 1. The hole will represent a blackbody.
   c) Monitor the object surface and the blackbody and set the emissivity correcting function of the thermal imager so that the temperature of the blackbody and that of the object surface are the same. The obtained emissivity will be the emissivity of the object surface.
   d) After this, no changes in the emissivity settings need to be made when measuring the same object type.

4. HYPOTHESIS

Inside the mill, the stones are crushed against each other and against the mill shell. This will lead to an increasing temperature inside the mill. The temperature inside the mill is higher than the temperature outside the mill and the heat will flow out through the mill (according to the second law of thermodynamics (Young et al., 2004)). If a crack appears in the mill, more heat will flow out through the crack opening than through the surrounding material. The increasing temperature around the crack due to the increasing heat flow will identify the damaged spot. In a similar way, if a coil damage has reduced the thickness of the mill, more heat will pass out there and the temperature will rise. A coil damage is when the linings have loosened from the mill shell and a combination of water and finer stone pieces seeps into the gap between the shell and the linings. The repeating movements, due to the mill rotation, of the water and stone combination in the area then lead to wear and thickness reduction of the mill shell. Not only will the cracks and material thickness reduction affect the surface temperature, but a different mill geometry and components like bolts and linings will also affect the heat flow. Moreover, all types of irregularities in the material, such as welded areas or blow holes from casting, will affect the surface temperature.

5. METHOD

During the measurement an NEC type TS9260 thermographic infrared imaging system with the following specifications was used for the monitoring of the mills (NEC TS9260, 2010-12-07):

- NEC 640 x 480 UFPA pixel detector
- Temperature range between -40°C and 500°C
- Thermal sensitivity (NETD) of 0.06°C (at 30°C and a sampling rate of 30 Hz)
• Accuracy of ± 2% (reading) or ± 2°C
• Frame rate of 30 frames/sec (Hz)

The measurements were taken with an passive approach. The mills were monitored from different views and the thermal images were analyzed for irregular thermal differences which could be related to material damage. Due to the mill rotation, it was hard to identify the critical spots found with an IR-camera in the reality on the mill. To solve this, an aluminium foil can be placed at a known position on the mill. The foil will be visible on the thermal images and can be used as a reference to find the critical spots while inspecting the mill. While taking the measurements, the camera had to be focused and kept stationary at the measurement position for several mill rotations in order to monitor the temperature differences as the mill rotates.

5. RESULTS

FIGURE 2 below shows the thermal mapping of the outlet head of an AG mill with a welded crack at a known position. The mill was monitored during its operation for several mill rotations. The left picture shows a crack-free part of the mill head and the right picture shows the portion with the welded crack. The numbers at the positions of the cursors indicate the surface temperature in °C. The increased temperature towards the centre of the mill in the picture to the right in FIGURE 2 indicates the welded crack. The position for the increased temperature was in agreement with the position of the welded crack.

FIGURE 2. Thermography images of an AG mill head, the picture to the left shows a normal crack-free portion of the mill head and the higher temperatures in the picture to the right indicates the welded crack.

In FIGURE 3, a thermal mapping of an SAG outlet mill head with a crack at a known position is shown. The mill was monitored during several mill rotations. The left picture shows a crack-free portion with normal temperatures and the right picture shows higher temperatures due to the crack. As for the AG mill, the position for the increased temperature was in agreement with the position of the crack.

FIGURE 3. Thermography images of an SAG mill head, the picture to the left shows the temperature field for a crack-free portion and the picture to the right shows higher temperatures due to the crack.
Another interesting result is shown in FIGURE 4. The pictures show the thermal mapping of the shell of an AG mill in operation monitored for several mill rotations. The picture to the left shows the normal temperature state in the mill and the picture to the right shows that something is leading to a higher surface temperature than normal. This is probably due to not completely fastened linings or a coil damage.

![FIGURE 4. Thermography images of an AG mill shell. Picture to the left shows the normal temperature state and picture to the right shows a portion where the temperature is higher than normal.]

6. DISCUSSION & CONCLUSIONS

FIGURE 2 and FIGURE 3 show the surface temperatures of mill parts where a crack is known to be located by the mill operators. The surface temperature difference between the damaged portion and the healthy portion is here approximately 1°C. The temperature on the inside mill surface is higher than the temperature on the outside mill surface. Because of this the measured temperature difference between the damaged portion and the unbroken portion is also higher inside the mill than outside the mill. In FIGURE 4, there is an outside surface temperature difference of approximately 0.5°C. In this part, there is most likely no crack, and a more probable explanation is that the linings in this part of the mill are not properly fastened to the mill shell. This will create a distance between the mill shell and the linings and more heat can pass out here. It is also possible that the loose linings have led to a coil damage which has reduced the mill shell thickness in the area.

When measuring the temperature of an object with a radiation thermometer, it is important to take into consideration the emissivity correction as well as the environmental conditions where the measurements will be performed. Infrared rays enter the thermal imager from the object undergoing measurement as well as all the other objects nearby. Therefore, in order to avoid this influence, a function of environment reflection correction, etc. is required. Moreover, when accurate data are required, it is necessary to minimize the influence by shortening the transmission route of the infrared ray, for example by shortening the distance between the object subjected to measurement and the thermal imager.

Based on the results presented in this article, it is likely that the infrared thermal camera can be a suitable method for health monitoring of rotating mining mills. The technique can be used to scan the mills rapidly for irregularities in the temperature and to find damaged areas. The suspected areas can then be inspected manually directly or during the next maintenance stop, depending on the location and the level of temperature difference in the suspicious zones. By repeating the inspections in a critical area, the material degradation can be monitored. If, for example, a crack is found, the crack propagation can be monitored. The most important cost factor for the mills is the downtime cost. By using a thermal imaging camera, the mills can be inspected and monitored remotely for damage without stopping the mills and costs can be saved. Based on the results for the inspection and monitoring, the maintenance activities can be more effectively planned and performed more precisely. This will not only optimize the maintenance activities, but also reduce the operation and maintenance costs.
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